MODERN ELECTRIC PRACTICE

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1. Steam-Boilers

CHAPTER I

INTRODUCTION

A history of the development of the steam-boiler from its earliest forms would be somewhat out of place in a book on modern electric practice; it is therefore considered best to limit this article to the description and illustration of the principal types of boilers in use in this country in electric light and power installations, giving at the same time some of the principal points to be considered by engineers in the choice of a boiler, and some model specifications.

The writer has not thought it incumbent upon him to confine himself to boilers alone, and he does not think the subject would be adequately dealt with if he did not touch upon such intimately allied matters as superheat, draught, the size of chimneys, the combustion of fuel, and mechanical stoking; while the manufacture and design of steam-boilers would form the subject of a separate treatise, and is not dealt with here.

It is proposed to briefly describe the principal types of boilers used in the electrical undertakings of to-day, and it is considered that the object of this article will be attained if boilers of the Lancashire, marine, dry-back, and locomotive types, and the water-tube boilers of the principal makers, are dealt with. These are the boilers which are generally met with in this country, while on the Continent and in America the installation of the water-tube boiler is almost universal.

The various parts of a boiler may be divided into three, as follows:—(1) The boiler proper, consisting of a vessel to contain the water and steam; (2) the furnace; (3) the setting.

HEATING SURFACE.—Those portions of the boiler which have water on one side and the hot gases produced in the combustion chamber on the other, form what is termed the "heating surface". The value of this heating surface in evaporating water varies with the type of boiler. It may be stated here that in this respect all boilers may be broadly divided into three classes, viz.:—(1) shell boilers; (2) fire-tube boilers; (3) water-tube boilers.

Shell Boilers.—Of the shell boilers we may mention the egg-ended boiler, the Cornish, and the Lancashire. An egg-ended boiler is a simple cylinder, with no internal flue, having spherical ends, and is externally fired, the flues being composed of brickwork. A Cornish boiler is also of cylindrical shape, but having one internal flue in the form of a tube. The grate is placed in the front end of the tube, and the boiler is internally fired. A Lancashire boiler is similar to the Cornish, but has two circular internal flues or tubes side by side.

The egg-ended boiler, except in some old-fashioned collieries, has passed out of use, and the Cornish boiler is only used in very small installations, where considerations of first cost outweigh the advantages of efficiency. Of shell boilers, the Lancashire type is most largely used. The heating surface of this boiler is partly internal and partly external, and will be understood by those who are not already familiar with it by reference to the description on page 6.

Fire-Tube Boilers.—Under this heading are classed locomotive, marine (wet-back), and dry-back boilers. The underfired multitubular boiler need not be considered, as it is rarely used.

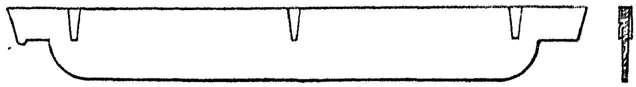


Fig. 1241.—Fire-Bar

The locomotive boiler has been an excellent servant for its own purpose, but for electrical undertakings it has not been largely used.

The marine boiler, sometimes called the Scotch boiler, has at sea proved itself an excellent steam generator, both reliable and economical, and it has been tried at several electric-light stations, notably at those of the St. Pancras Vestry in London and at the electric-lighting station of the Bradford Corporation.

In each of the above three types the gases pass from the combustion chamber through tubes surrounded by water, and thence to the smoke-box and to the flue or chimney. With the water-tube boiler the reverse is the case. The water is contained in the tubes, which are arranged in a group or series of groups in the combustion chamber itself, and these groups of tubes form the principal part of the heating surface.

THE FURNACE.—The grate is formed of cast-iron fire-bars of varying length according to the size of the grate, each bar being generally from 3 feet to 3 feet 6 inches long. The most general form of bar is illustrated in fig. 1241, from which it will be seen that there are projections at each end to keep the bars apart, the dimensions of which regulate the space for admission of air. The space varies according to the character of the fuel and the strength of the draught; but it may be stated generally that for coke and coals the total air-space should be from .3 to .25 of the total grate area, and for turf and wood chips from .2 to .13 of the total grate area. The bars should be cast from hard scrap; if they are made of too soft metal, they will soon burn away and require renewal. They are supported on cross bearers, which are fixed to the sides of the furnace, and allowance must be made for expansion of the bars as they become heated. The allowance generally amounts to $\frac{1}{2}$ inch per length of bar.

The area of the grate varies with the type of boiler, the desired rate of combustion, the character of the fuel, and of course the amount of steam to be generated.

The length of the grate is limited, if hand firing be employed, by the distance the fireman is able to throw the coal, so as to keep the back of the grate covered, and to avoid undue admission of cold air at that part through holes burned in the fuel. As a matter of practice, it is found inexpedient to have a grate more than 7 feet in length if, as stated above, hand firing be employed.

THE COMBUSTION CHAMBER.—In shell and fire-tube boilers the combustion chamber has metal walls, with the water contained in the boiler on the opposite side, and therefore forms part of the heating surface as previously defined. In water-tube boilers it is usually formed of brickwork; the fire-bricks with which it is lined should be of the best quality obtainable, so as to avoid unnecessary renewals and repairs.

The fuel is introduced through doors, which are usually fitted with an inner perforated plate or plates, which serve the double purpose of keeping the outer door cool and of breaking up the air admitted through openings in the main door into small jets. This very much conduces to perfect combustion of the fuel.

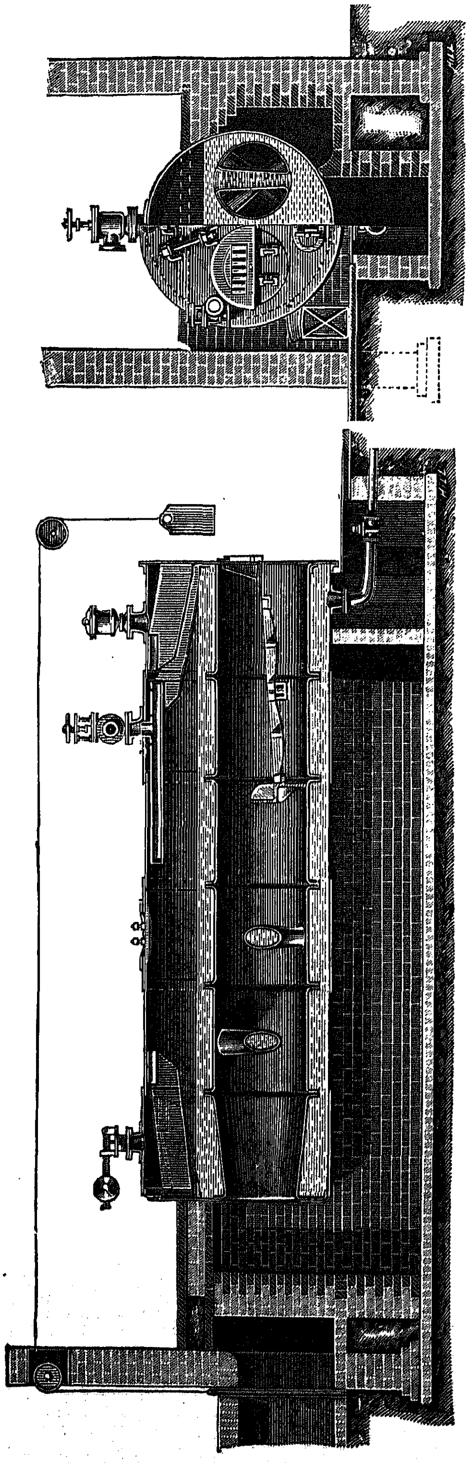
SETTING.—The setting, which is the brickwork portion of the boiler, includes the flues, which may be described as conduits for conveying the gases from the combustion chamber to the chimney. With the Lancashire boiler they play an important part, conducting the gases from the internal tubes or flues, underneath the shell of the boiler, and round the side wings, so that the outside of the shell may be used as heating surface. This will be understood by reference to figs. 1247 and 1248 on page 12.

Where the type of boiler permits, it is desirable to so design the flues that the current of gases should be as far as possible at right angles to the heating surface. It is an elementary principle in designing flues to avoid sharp bends and ensure a smooth surface.

CHAPTER II

TYPES OF BOILERS

The Lancashire Boiler.—The Lancashire boiler has been for many years a great favourite with steam users in this country, in mills and factories, where pressures not greatly exceeding 100 lbs. per square inch obtain, and where large units of boiler power are not adopted. It is economical, and, for moderate pressures, cheap in first cost, while up to a certain point it will stand a good amount of rough usage; having a large area of water surface, and containing a considerable quantity of water, it produces a good and steady supply of steam. It is a horizontal boiler of cylindrical form, with flat ends, having two internal parallel tubes or flues extending from one end of the shell to the other (fig. 1242). The shell is composed of courses of plates, each about 3 or 4 feet wide, and so arranged that each course is alternately an outer and an inner belt. In modern practice each course is made of one plate. The end-plates are made from one plate also; the front



Cross Section

Longitudinal Section

Fig. 1242.—Lancashire Boiler

plate is attached to the shell by an angle ring, and the back plate is flanged to join the shell. These end-plates are stayed to the shell by what are termed gusset-stays. There are usually five of these above the flues in boilers constructed for pressures up to 120 lbs., and six for higher pressures. Below the flues there are three at the back and two at the front. The reason for the smaller number at the front is to allow room for the mud-hole. The sectional area of the gusset-stays must be designed to resist the required pressure. Half the gusset-stays should be riveted to the first course of the shell, and the remainder to the second course. The angles on the end-plates, to which the gussets are attached, should not be brought close down to the flues, but should terminate from 8 to 10 inches away, so as to allow the end-plates to "breathe", as it is termed. If too rigidly attached to the shell, grooving of the flanges of the flue will take place in a very short time after the boiler is set to work. In addition to the gusset-stays, long tie-rods, connecting the front and back end-plates, are fitted by many makers, but it is doubtful whether these are of much service, and the modern tendency seems to be to dispense with them.

The plates forming the flue are generally flanged outwards at the ends, and connected together as shown in fig. 1243; a ring being inserted between the flanges to enable the joint to be caulked. This form of joint is known as the "Adamson", from the name of the original inventor. The advantage of this form of joint is, that while adding greatly to the resistance of the flue to collapse, the heads of the rivets connecting the plates are removed from the heat of the gases from the fire.

Galloway Tubes.—In order to increase the heating surface of the boiler, and also, it is claimed, to increase the somewhat defective circulation, and so help to equalize the temperature of the boiler and prevent unequal expansion, it is a common practice to introduce into the furnace tube a number of what are called Galloway tubes, one of which is illustrated in fig. 1244. Manholes are also provided, one on the top of the shell and the other at the bottom of the front end-plate, below and between the furnace mouths.

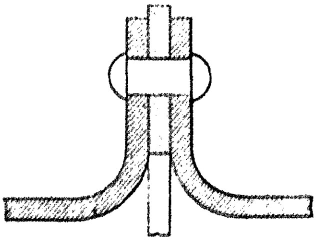


Fig. 1243.—Adamson Ring-Joint

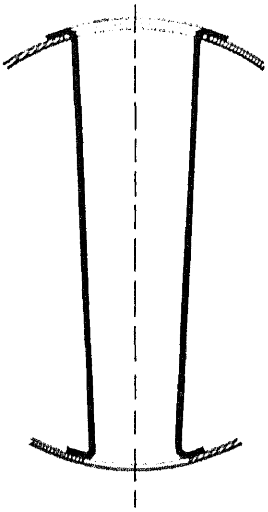


Fig. 1244.—Galloway Tube

Manhole.—The manhole in the front plate is generally oval. A compensating ring is riveted round the hole for strengthening purposes (fig. 1245).

Furnace.—The furnace has been described above, and will be understood from the illustration. A fire-brick bridge is placed at the end of the grate to keep the ashes from falling over the end of the bars. The fire-bricks are held in place by a casting.

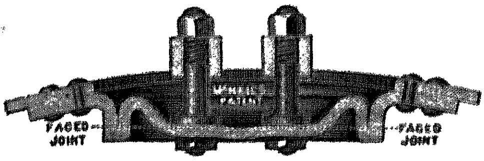


Fig. 1245.—Manhole Door

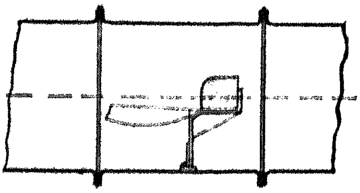


Fig. 1246.—Furnace Bridge

Bridge.—The height of the bridge is a matter of opinion; but the writer's experience is that the bridge should only be just high enough to prevent the fuel from falling over, and to offer no obstruction to the course of the gases (fig. 1246).

With regard to mountings, *i.e.* valves, gauges, &c., as these apply to all classes of boilers, we shall deal with them separately.

A model specification is given below for a Lancashire boiler 30 feet long × 8 feet diameter, 160 lbs. working pressure:—

SPECIFICATION OF A LANCASHIRE BOILER FOR A WORKING PRESSURE OF 160 LBS. PER SQUARE INCH

Dimensions and Particulars.

Length of boiler	30 feet
Diameter of shell	8 „
Furnace-tubes—	
Diameter of first nine rings to be	3 feet 2 inches.
„ tenth ring to be tapered to	2 „ 8 „
„ eleventh ring to be parallel	2 „ 8 „
„ rivet holes, shell, longitudinal seams ...	$\frac{13}{16}$ inch.
„ „ „ circular seams	1 „
„ „ furnace-tubes	$\frac{13}{16}$ „
„ „ gusset-stay	1 „
Thickness of plates, shell	$\frac{3}{4}$ „
„ „ furnace-tubes	$\frac{9}{16}$ and $\frac{5}{8}$ „
„ „ ends	$\frac{3}{4}$ „
„ „ butt-straps	$\frac{5}{8}$ „
„ „ gusset-stay plates	$\frac{3}{4}$ „

Shell angle-rings, 5½ × 5½ × $\frac{13}{16}$ inches, double-riveted.
Furnace-tube, flanged rings, Adamson.
Gusset-stay angle-bars, 3½ × 3½ × $\frac{5}{8}$ inches, single-riveted.
Gusset-stay angle-bars to shell to be set to suit gusset-plates as required.

Weight of plates, shell	30.6 lbs.	per square foot.
„ „ furnace-tubes ...	23 and 25.5 „	„
„ „ ends	30.6 „	„
„ „ butt-straps ...	25.5 „	„
„ „ gusset-plates ...	30.6 „	„

Note.—The diameter of the shells and furnace-tubes to be measured internally, that of the shell being measured inside the outer rings of plates.

Materials. Plates. The shell-plates, butt-straps, and end-plates to be made of Siemens-Martin steel, having a tenacity not exceeding 30 tons or less than 26 tons per square inch, and to afford an elongation of at least 20 per cent in a test-piece 10 inches long.

Angle-Rings and Furnace-Tubes. To be made of Siemens-Martin steel, having a tenacity not exceeding 28 tons nor less than 24 tons per square inch, and an elongation of 25 per cent in a test-piece 10 inches long.

Brands. Each plate to be branded and numbered, care being taken during construction that the numbers and brands will be in a prominent position. The tensile strength and ductility of each plate to be stated on the record of tests furnished by the manufacturer.

Test-Strips. Strips from the plates of shell, furnace-tubes, and angle-bars are to be capable of being bent cold to a radius of one-and-a-half times the thickness of the plate without fracture, after having been heated to a cherry-red, and plunged into water of 80° F. The welding properties of strips taken from the angle-bars and furnace-plates are also to be ascertained by actual trial, and generally the materials are to be capable of withstanding the various tests to the satisfaction of the inspector appointed for the purpose.

Rivets. To be of specially-selected rivet steel, the tensile strength of which shall be from 24 to 27 tons per square inch, and the elongation in a length of 8 inches to be not less than 25 per cent. Samples of the rivet steel are to be submitted to bending, breaking, and flattening tests.

CONSTRUCTION

Shell. The shell to consist of eight parallel belts of plating of equal width alternately in and out, the first belt of plates being an outer one. Each belt to be formed of one plate, and all to be made perfectly cylindrical.

Seams of Rivets. The longitudinal seams of shell to be butt-jointed, with cover-straps inside and outside, and to have three rows of rivets at each side of the butt, or six rows in all, the calculated strength of the joint being at least 85 per cent of the solid plate. These seams to be at the upper side of the boiler, right and left sides alternately, and to be arranged so as to clear the brickwork and mountings. The circular seams to be double-riveted with lap-joints.

Butt-Straps. The butt-straps to be cut to the required sizes from plates of the same quality as that specified for the shell, and care must be taken that the straps in the direction of their width will be lengthwise of the plates they are cut from. The ends of the butt-straps, where they tuck under the ring seams, may be thinned by planing or by forging. If the strips are heated for forging or bending they should be afterwards annealed.

End-Plates. Each end-plate to be rolled in one piece. The front end to be attached to shell by a solid-welded angle steel ring, which shall be fixed externally, the back end-plate being flanged to meet the shell. The edges of both plates and angle-rings to be neatly turned up, and the holes for flues to be cut out by machine.

Stays. The steam-space portion of end-plates to be strengthened by means of six gusset-plate stays, the total section of which must not be less than 42 square inches when measured at their weakest parts. The bottom rivets in all the gusset-stay angles on the end-plates must be concentric with the rivets joining the furnace-tubes to the end-plates, and the distance between the bottom rivets of stays and the rivets joining the internal flues to be 10 inches. Each end-plate to be strengthened below the furnace-tubes. Back end-plate by means of three gusset-plate stays, and the front end-plate by two gusset-stays. All the gussets to be united to shell and ends by double angle-bars.

Furnace-Tubes. The furnace-tubes to be truly cylindrical, the longitudinal joints being welded. Each tube to consist of not less than eleven belts of plating, united to each other by Adamson's flanged seams, having solid caulking-rings of same thickness as plates between each. The first and last belt of each furnace-tube to be increased to ⅝ inch in thickness, so as to allow of strengthening collars being dispensed with, the thicker plates being flanged for attachment to ends in the ordinary manner. Care must be taken to prevent the circular seams of flues from falling in line with each other, or with those of the shell.

Alternative Price to be quoted if Galloway Tubes dispensed with. Each furnace-tube to be fitted with five of Galloway's steel cross-pipes, the first of which shall be placed vertically in the fifth belt of plates, the others being fitted in the consecutive belts, and set towards right and left sides alternately at an angle of 30° from the vertical. Each furnace-tube to taper at the last belt but one from the back end, the last belt in each being made parallel to the diameters specified.

Rivet Holes. All rivet holes to be drilled full size through the solid plates after they are bent, and wherever practicable, the plates, angles, and flanges are to be drilled in position. The holes are afterwards to be slightly countersunk at the rivet heads, and care must be taken to remove the burr from between the plates. Should any of the holes be unfair when the plates are drawn up, they must be reamered out before riveting.

Riveting. To be done wherever practicable by machine.

Edges of Plates The edges of all the plates and butt-straps throughout the shell to be planed, the front and back end-plates to be turned at the furnace-tube openings, and the outer edges of the seams of the furnace-tubes to be faced in machine. The distance between the centre of rivet holes and edges of plates to be equal to one-and-a-half times the diameter of the rivets.

Caulking. The seams throughout to be caulked or fullered internally and externally.

Drawings. The makers will be required to furnish two sets of tracings of the boiler, with figured dimensions, showing general arrangements; details of the stays and riveting are also to be submitted for approval.

Flanging and Welding. These to be done with the special tools and appliances for such purposes, and all plates which have been worked in the fire are afterwards to be annealed when completed.

Inspections during Construction. The makers will be required to send notice to   at the following stages of construction:—

1st. When the plates are in hand, and before they are bent.
2nd. When they are being drilled, and before riveting is begun.
3rd. When they are in process of being riveted.
4th. When the boiler is completed and ready for being tested.

In addition to these, the boiler must be open to examination by the inspector appointed at any time during construction.

Tests. The boiler to be tested to 240 lbs. per square inch by water-pressure in the maker's yard, and to sustain the same to the satisfaction of the inspector appointed for the purpose. The boiler must also be tested when it has been placed on its seating with all the mountings in position except the safety-valves, it being understood that these valves will be tested and guaranteed by the makers. The date of make and pressure to which the boiler is tested to be legibly stamped on the front end-plate.

Mountings and Fittings

Steam Stop-Valve. One steam stop-valve, 6 inches diameter, having wrought-iron cross-head and wrought-iron pillars.

Safety-Valves. One dead-weight safety-valve, 2½ inches diameter, fig.   in   list, and one combined 2½ inches diameter high-pressure and low-water safety-valve, fig.   in   list.

Each valve to be tested for a pressure of 160 lbs. per square inch.

Feed-Valve. One accessible feed-regulating and back-pressure valve, 2½ inches diameter, fig.   in   list, to be attached on top of shell or front end as required, and to be fitted with an internal pipe properly supported, and arranged to discharge horizontally about 3 inches above the level of the furnace crowns, and at least 12 feet from front end.

Suitable hand-wheel and gear to be provided for operating feed-valve from foot-plate.

Water-Gauges. Two sets of asbestos-packed glass-tube gauges, ¾ inch diameter, with safety-shields, fig.   in   list, having solid-drawn copper drain-pipes of suitable length. The water-level to be 5 inches above the furnace crowns when it is visible at bottom of glass tubes.

Water-Level Pointer. One brass water-level pointer of neat design to be fixed at a height of about 10 inches above the furnace crowns.

Steam-Gauge. One 12-inch steam-pressure gauge, fig.   in   list, with red mark at 160 lbs., graduated to 320 lbs. per square inch, fitted with tap and syphon complete. To facilitate the testing of gauge, the tap must be made suitable for opening to the atmosphere or boiler as required.

Blow-off Cock and Bend. One 2½-inch asbestos-packed blow-off, fig. in list, entirely of gun-metal, and having the gland constructed so as to prevent the key being removed until the cock has been closed. One cast-steel elbow-pipe of approved form, having an internal diameter of 6 inches at its upper end, and tapering down for connection to cock. The bend-pipe to be strongly ribbed, and the metal to be not less than ⅞ inch in thickness at the largest diameter.

Antipriming Pipe. One perforated cast-iron pipe to be placed horizontally in steam space, and connected to steam stop-valve. The pipe to be carried close to the crown of shell by means of suspenders attached to a small riveted bracket at each end. The united area of the perforations in this pipe to be about 25 per cent in excess of the area of the stop-valve.

Fusible Plugs. One low-water fusible plug to be fitted into the crown-plate of each furnace directly above the centre of fire-grate. Spare caps to be supplied. Makers to guarantee the suitability of the plugs for the working pressure.

Manholes. One of M'Neill's flanged manhole-rings, with patent embossed door, to be fitted on top of shell, and one at front end below furnaces, each having a clear opening of not less than 15 × 11 inches, all being proportioned for a working pressure of 160 lbs. per square inch.

Branches for Mountings. The branches to be of wrought steel, having the upper flanges accurately faced for jointing. The bottom flanges to be of the same radius as boiler, and the branches are to be riveted to shell at the positions required for steam-pipe and other connections.

Furnace Mouthpiece. One wrought-iron or steel mouthpiece for each furnace, finished off with neat moulding outside, and fitted internally with a cast-iron arch-piece surrounding the fire-door for carrying a fire-clay lining to protect the furnace front. The cast-iron door to be fitted with a sliding ventilator, affording an area of about 20 square inches, as well as with an internal perforated baffle-plate.

Dead-Plates and Fire-Grates. Two dead-plates and a complete set of fire-bars in two lengths of 3 feet 6 inches each, the total length of grate surface being 7 feet. The dead-plates as well as the bearers for fire-grate to be carried by wrought-iron brackets riveted to the sides of furnaces.

Flue Doors. Two side-flue doors and frames sufficiently large to afford easy access to external side-flues.

Damper and Frame. Two damper-plates and frames with all fittings necessary for operating damper from foot-plates.

Half-moon dampers adjustable and removable to be fitted under each furnace mouthpiece.

Evaporation.—The following table may be found useful, giving the maximum economical evaporations of several sizes of Lancashire boilers with feed-water at 150° F. with good coal:—

Size.	Heating Surface in sq. ft.	Galloway Tubes.	Diameter of Furnaces.	Evaporation in lbs. per hour.
28 ft. × 7 ft.	852	8	2 ft. 9½ ins.	5000
30 „ × 7 „	920	10	2 „ 9½ „	5500
28 „ × 7 „ 6 ins.	920	8	3 „ 0½ „	5500
30 „ × 7 „ 6 „	1000	10	3 „ 0½ „	6000
30 „ × 8 „	1068	10	3 „ 3 „	6600

Foundations and Setting.—Upon the brickwork setting of a Lancashire boiler depends not only its capacity, but largely its efficiency. A concrete bed should be provided *at least* 6 inches thick, depending, of course, upon the nature of the subsoil. The boiler is usually seated upon special fire-brick blocks. The object of these is, not only to provide a secure seat, but also to prevent moisture accumulating in contact with the boiler plates, thus setting up external corrosion. The boiler should be set with a slight fall, say, 1 to 1½ inch, towards the front to drain it completely. The drainage of the foundations should be given especial care for the reason above stated.

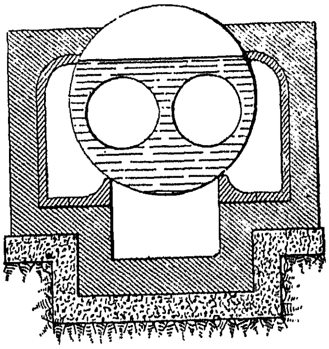


Fig. 1247.—Cross Section of Lancashire Boiler

Flues.—The flues should be carefully proportioned to allow ample room for the passage of the gases, and for a man to get in for inspection and cleaning purposes. The side flues are generally made 9 to 10 inches wide, and are covered with special fire-clay flue covers, as illustrated above. The bottom flue should not be less than 2 feet deep, measuring from the bottom of the boiler, and about as wide as half the diameter of the boiler. The general practice is to pass the gases through the bottom flue, towards the end of which is a division wall, which splits the gases as they rise to the side flues. The divided streams unite again in the main flue, and pass either through an economizer or direct to the chimney (fig. 1247 and 1248).

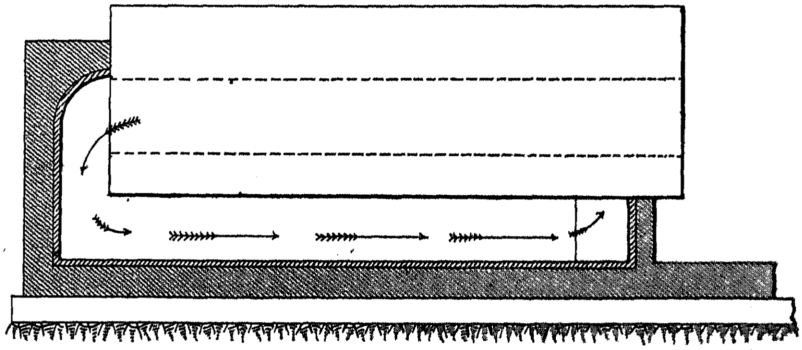


Fig. 1248.—Longitudinal Section of Lancashire Boiler

Brickwork.—Mortar should not be used for setting the boiler on the seating blocks. The bricks should be set with a mixture of fire-clay and water, and made very thin. If the fire-clay is laid on thickly with a trowel, the brickwork is apt to crack with the heat.

Dampers.—Dampers are placed, one in each side flue, and are usually actuated from the front by means of steel wires or chains.

The Galloway Boiler.—The Galloway boiler is illustrated in fig. 1249. Like the Lancashire, which it resembles in outward appearance, it consists of a cylindrical shell, and at the front end it has two circular flue-tubes, which on the far side of the flame bridges unite into one flue-tube. A number of Galloway tubes are fitted into this flue-tube to break up the gases, and increase the heating surface and the circulation. Side pockets are also made in this flue for the same reasons.

Messrs. Galloway arrange for the setting of this boiler to be somewhat different to that of the Lancashire boiler, and the plan they adopt is that the gases, after leaving the furnaces and passing through the Galloway flue, divide at the back end, and return along the two side flues to the front, where they descend, reunite, and pass under the bottom of the boiler to the main flue.

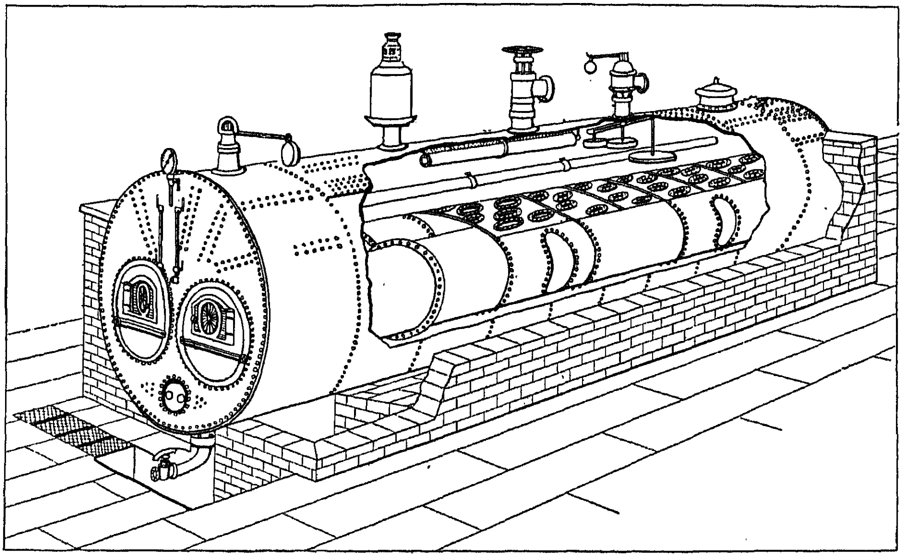


Fig. 1249.—Galloway Boiler

The Dry-Back Boiler.—This is a type of boiler which is now very considerably used in electric installations, and in other situations on land. It is a modification of the well-known "Scotch" boiler, or marine return tubular boiler, which has been used on board ships for so many years. In the ordinary "Scotch" marine boiler the combustion chamber is situated within the shell of the boiler, and has a water space all round its sides and also between its back-plate and the rear end-plate of the boiler, whereas in the boiler now under consideration the combustion chamber is external to the boiler, and has its sides and back formed of fire-brick. Hence the name "Dry-Back" boiler, by which it was first known on the Clyde, to distinguish it from the ordinary marine return tubular boiler, which is sometimes in that district called a "Wet-Back" boiler.[1]

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[1] While the appellation "dry-back" accurately describes this type of boiler, it is also known under other names, such as the "economic" boiler, the "semi-marine" boiler, and sometimes simply as a "marine" boiler.

In the dry-back boiler the flue or flues go right through from end to end of the shell, and above them are situated the return tubes, which are generally about 3 inches in external diameter, and which also extend from front to back the whole length of the boiler. Care is taken in the design that the upper row of tubes is placed at such a height in the boiler as will leave room for a large steam space above the surface of the water when at the working level.

The gases pass from the furnaces through the flues and into the combustion chamber at the back end, where they get intimately mixed together, and as the fire-brick lining of this chamber becomes raised to a high temperature, in the course of working, the combustion of the products of the fuel is very thoroughly accomplished. From the chamber the hot gases next pass through the small return tubes to the front end of the boiler, where they enter the smoke-box. This is an iron casing fixed on the front of the boiler, and provided with doors and cleaning holes for the sweeping of the tubes and for the removal of soot and dust.

For fixed situations, and in order to realize the best evaporative results, dry-back boilers are usually set in a brickwork setting, as shown in fig. 1250, which shows a boiler with the brickwork in course of erection; the wall at one side being left incomplete allows the boiler to be seen resting on its cast-iron cradles or supports.

The smoke-box is generally divided into two parts by means of a vertical division plate at its middle, and the gases are thus compelled to leave it in two portions, one going to one side of the boiler and the other to the opposite side, passing down between it and the brick side-walls, thence under and along the bottom of the boiler towards the back, where they enter an underground flue leading to the chimney. It will be seen that the hot gases are thus compelled to traverse twice within the shell of the boiler, and once along its outer surface, an arrangement which gives a very large amount of wetted surface in useful contact with the heating gases, while at the same time there is only a very small area of brickwork for the absorption and subsequent loss of heat by radiation.

Fig. 1251 shows two dry-back boilers for an electric installation, set side by side and forming part of a range of boilers to be set in brickwork in the manner described. The dry-back boiler, however, is very frequently installed without any brickwork surrounding it, in which case it stands clear, resting on its cast-iron cradles and available for inspection all round its exterior, like an ordinary return tubular marine boiler on board a ship. With this arrangement the gases may be led away from the smoke-box directly into an iron chimney, or by means of a down-take they may be led into an underground flue and thence to a chimney, or, as shown in fig. 1252, they may be collected in an overhead flue serving a range of boilers and conveying the gases to a large brick chimney-stalk.

The dry-back boiler with a good draught gives good results, and it has been proved to be economical in working, and at the same time it is claimed that the cost for repairs and upkeep is a low one.

Mechanical draft, either forced or induced, can be as easily applied

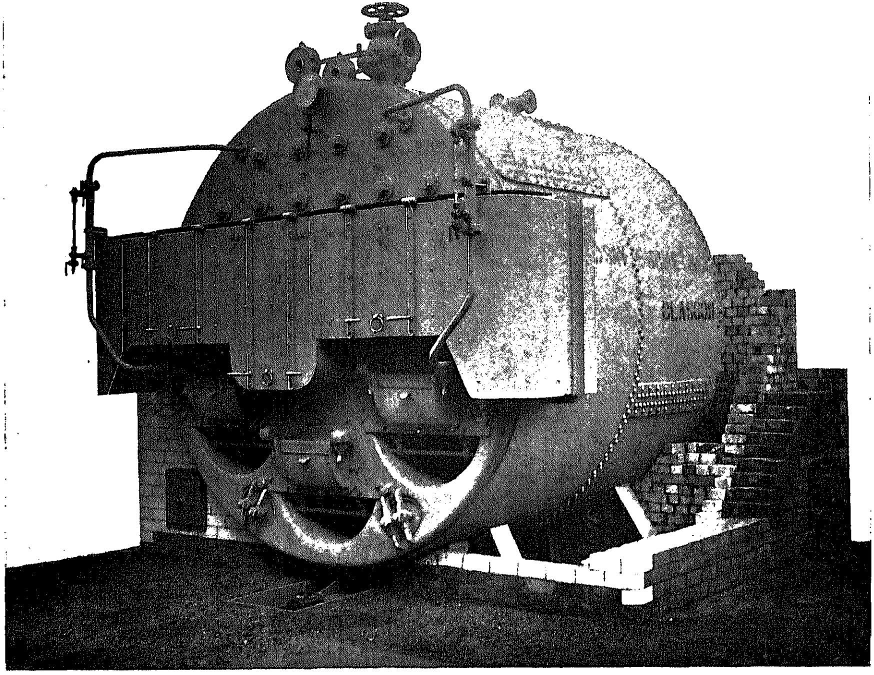

Fig. 1250.—Dry-back Boiler, showing Brickwork setting

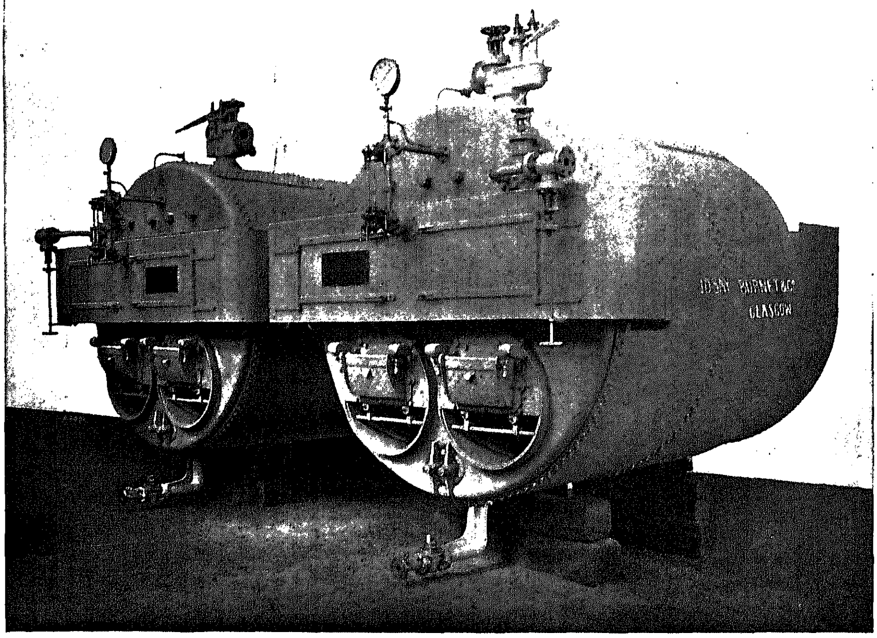

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Fig. 1251.—Dry-back Boilers for Electric Installation

to it as to the ordinary marine boiler, and under these conditions its evaporative duty can be highly increased.

Locomotive Boiler (Stationary Type).—This class of boiler is so seldom used in large electric-light stations, that for the purposes of this work it hardly seems necessary to describe it; still, in some smaller installations, especially when combined with an undertype engine, it has been found useful, especially for temporary work.

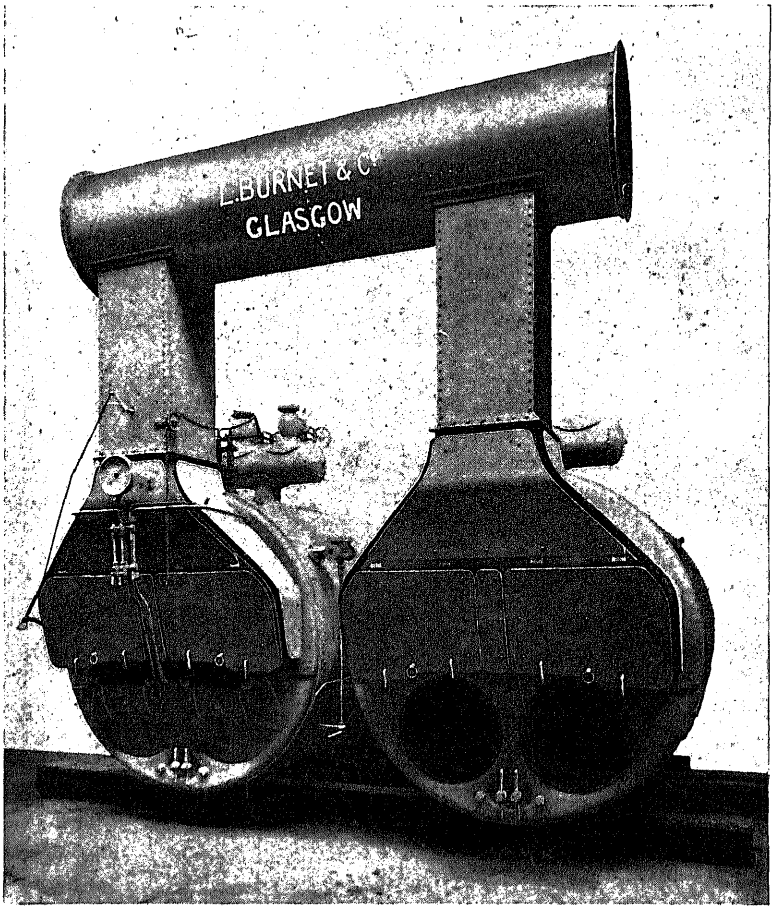


Fig. 1252.—Dry-Back Boilers with Overhead Flue

This boiler (fig. 1253) consists of an outer shell of cylindrical shape through which a number of tubes are passed, and of a shell surrounding the fire-box, which is necessarily of a similar shape to the latter. Part of the fire-box is a tube-plate; the other tube-plate forms practically the end of the pressure part of the cylindrical shell, and separates it from the smoke-box. Each end of the fire-tubes is expanded into these tube-plates. Some proportion of them is screwed at both ends to act as stay-tubes. The fire-box forms also the combustion chamber, the sides and crown of which are surrounded by water, the grate being at the bottom of the chamber. It will be seen that the gases pass from the chamber through the fire-tubes to the smoke-box, and thence to the chimney. The crown of the fire-box is usually fitted with either one or two fusible plugs, and, being a flat surface, is stayed with dog-stays. The boiler shell is usually covered with felt or silicate cotton, held together by a layer of wood, which in turn is covered with sheet-steel. The ratio of heating surface to grate surface varies from 25 to 1, to 40 to 1, and with strong draft as much as 40 lbs. of coal can be burned per square foot of grate.

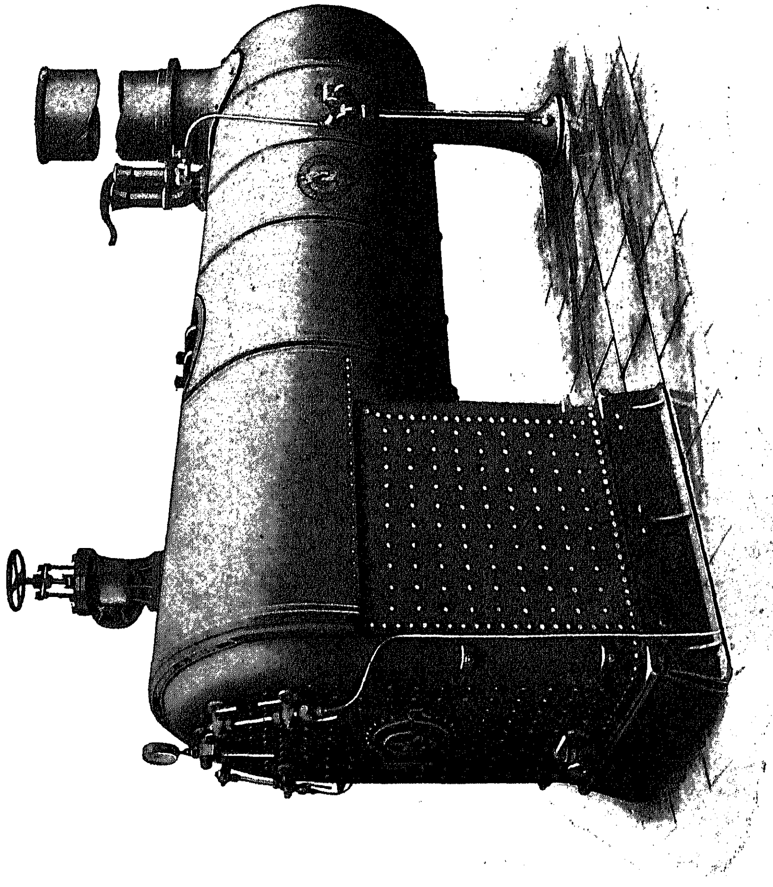


Fig. 1253.—Locomotive Boiler

The internal diameter of the tubes in all cases is 2¾ inches.

For calculating the evaporative capacity of a locomotive boiler, with

Total Heating Surface in sq. ft.	Grate Area in sq. ft.	Number of Tubes.	Length of Tubes.	Length of Barrel.	Diameter of Barrel.	Height of Chimney.
			ins. ft. ins.	ft. ins.	ft. ins.	ft. ins.
805.7	21	86	2¾ × 11 6¼	20 1½	4 8	22 5
642	16	80	,, × 10 0	17 6	4 6	22 2
496.5	13	70	,, × —	15 5	4 6	15 1½
409	10	61	,, × —	14 10½	4 0	16 9
322	10.8	50	,, × —	13 11	—	14 4
249	9.6	39	,, × 7 4$\frac{13}{16}$	13 3½	3 3⅞	13 6½
189.3	8	32	,, × —	12 0½	3 0	12 10
143.4	6.4	24	,, × —	11 6⅝	2 10½	13 0

good natural draught and best Welsh coal, 5 lbs. of water per square foot of heating surface can be allowed. A boiler on a running locomotive does considerably more than this.

CHAPTER III

WATER-TUBE BOILERS

Before describing the principal water-tube boilers, it will be well to consider the advantages which are claimed for this type, and which presumably have led to their very extensive adoption in lighting and traction plants in this country, abroad, and almost universally on the continent of America.

Circulation.—The chief claim made for it is that a properly-arranged water-tube boiler has a better circulation of the water than is possible with a boiler of the shell type. A good circulation increases the efficiency of the boiler in two ways: it enables the water to take up more heat in a given time, and decreases the waste of heat occasioned by priming.

The good circulation of well-designed water-tube boilers ensures the necessary uniformity of temperature; and moreover, it is possible to so arrange the boiler that, while steam is raised at a very quick rate of speed to the highest pressures, the various parts are free to expand and contract in every direction without the use of any stays whatever.

Safety.—Again, in a water-tube boiler the water space can be divided into sections, and so arranged that, should any portion fail, no general explosion can occur, the destructive effects being simply confined to the escape of the contents; owing to the small diameters of the parts, it is possible not only to use high pressures while allowing a very high factor of safety, but also to employ **thin heating surfaces** compared with the thick plates of ordinary boilers. This is an advantage which must lead to efficiency. The heating surface can be arranged in such a manner as to break up the gases and extract the maximum of available heat therefrom, and by this means the temperature of the gases at the chimney is reduced to the lowest possible point necessary to create sufficient draught to burn the fuel.

Combustion.—It is also claimed that it is possible to obtain better combustion than with some other types. As the walls of the combustion chamber are in most cases lined with fire-brick, a very high temperature is maintained, which ensures perfect combustion before any of the heat is abstracted. This arrangement also allows fuel of a very inferior quality to be used with efficiency.

Small Space.—Compared with shell boilers a water-tube boiler occupies very little ground space, power for power; thus a capital saving is made in the cost of land and buildings.

We give here an extract from a paper read by Mr. A. B. Mountain, electrical engineer of Huddersfield, before the Municipal Electrical Engineers. At the Huddersfield station both Lancashire and water-tube boilers are installed:—

Type of Boiler.	Floor Space Occupied in Feet.	Water Capacity in Lbs.	Steam Capacity in Cubic Feet.	Lbs. of Water Evaporated from and at 212° F. per lb. of Coal Consumed.
Water-tube	180	17,500	200	11.5
Lancashire	360	43,000	200	10.0
Lancashire, with economizer	410	76,000	260	12.0
Locomotive...	150	22,000	200	11.0
Dry-back marine	160	25,000	220	11.0
Marine	150	17,000	180	11.0

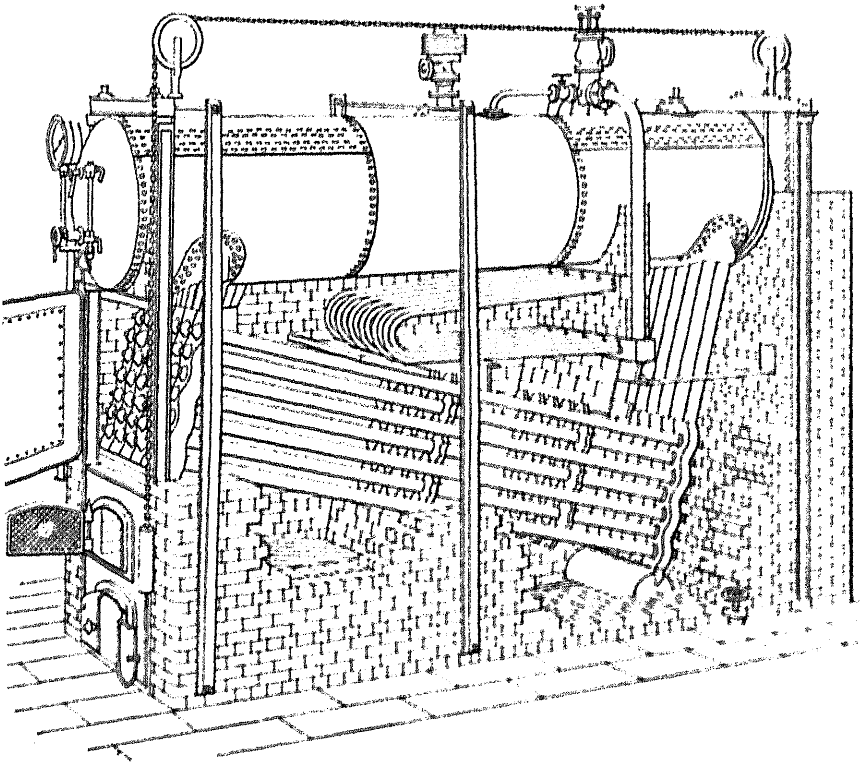
Accessibility.—It is also claimed that a properly-designed water-tube boiler is more accessible for cleaning and repairs, both inside and out, than one of the shell type; and moreover, it is more easy for a foreman to supervise the cleaning without trouble to himself, and therefore it is better done, than in the case of the shell boiler, where the foreman, who may be an elderly man, or may dislike creeping through a narrow manhole, may trust too much to the word of the workman.

Repairs.—Repairs in a water-tube boiler are of the simplest description, and practically always consist in the replacement of a tube, which can be effected by any mechanic; it follows that as a defective part is replaced by an entirely new one, the boiler itself is not weakened, and thus there is no necessity for reducing the steam pressure as time goes on, as is the case with all boilers of the shell type, because any repairs which it is necessary to effect to the latter cannot, from their very nature, be as strong as the original boiler structure.

Babcock and Wilcox Water-Tube Boiler.—*Construction.*—This boiler (fig. 1254) is composed of mild steel tubes, placed in an inclined position and connected with each other, and with a horizontal steam and water drum.

vertical passages at each end, while a mud-drum is connected to the r and lowest point in the boiler.

Tubes and Headers.—The end connections or "headers" are forged mild steel in one piece for each vertical row of tubes, and are of h a form that the tubes are "staggered" (or so placed that each row ıes over the spaces in the previous row). The holes are accurately d, and the tubes fixed therein by an expander. The sections thus ned are connected with the drum and with the mud-drum also by tubes expanded into bored holes, doing away with all bolts, and ng a clear passage-way between the several parts. The openings leaning, opposite the ends of each tube, are closed by mild-steel -hole plates, the joints of which are made by milling the surfaces curate metallic contact; the plates are held in place by mild-steel d clamps and bolts.



1254.—Babcock & Wilcox Water-Tube Boiler, fitted with Superheater (Integral with Boiler) for Stationary Purposes

rums.—The steam and water drums are made of mild steel and le-riveted. They can be made for any desired working pressure, are tested hydraulically to at least 50 per cent above the working ıre.

his boiler is suspended entirely independently of the brickwork, from ght-iron girders resting on iron columns. This avoids any straining of the boiler from unequal expansion between it and its enclosing walls, and permits the brickwork to be repaired or removed, if necessary, without in any way disturbing the boiler.

The furnace is placed under the front and higher ends of the tubes, and the products of combustion pass up between the tubes, down through them again, then once more up through the spaces between the tubes, and off to the chimney. The water inside the tubes, as it is heated, tends to rise towards the higher end, and the mingled column of steam and water, being of less specific gravity than the solid water at the back end of the boiler, rises through the vertical passages into the drum above the tubes (fig. 1255); here the steam separates from the water, and the latter flows back to the rear, and down again through the tubes in a continuous circulation. As the passages are all large and free, this circulation is very rapid, sweeping away the steam as fast as formed, and supplying its place with water, absorbing the heat of the fire to the best advantage, causing a thorough commingling of the water throughout the boiler, and a consequent equal temperature. The rapid circulation also prevents to a great degree the formation of deposits or incrustation upon the heating surfaces, sweeping them away and depositing them in the mud-drum, whence they are blown out.

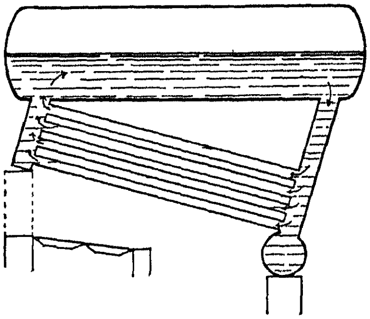


Fig. 1255

The steam is taken out at the top of the steam-drum near the back end of the boiler after it has thoroughly separated from the water.

The Babcock & Wilcox Marine Water-Tube Boiler.—The construction of the Babcock & Wilcox marine boiler (fig. 1256) is on the same lines, as far as the principle of water circulation, accessibility for cleaning, safety, and efficiency are concerned, as the land-type boiler of the same makers.

There are two designs of the marine type, one having medium-sized tubes, the other having large tubes, the selection of the type depending upon conditions of space and weight, &c.

As in the land type the boiler is constructed wholly of wrought steel, there being no cast metal of any description subject to pressure. It consists in the main of an arrangement of inclined tubes forming the bulk of the heating surface, sinuous boxes or headers to which the tubes are attached, a horizontal steam-and-water drum, a mud-drum, and a furnace of large capacity immediately beneath the inclined tubes, the relative positions of which are shown by the illustration on page 21.

The inclined tubes are grouped in vertical sections, and to ensure a continuous circulation in one direction are placed at an inclination of 15 degrees from the horizontal. The tubes are so arranged as to break up and ensure efficient contact with the products of combustion.

By distributing the surface into sectional elements all danger from unequal expansion, due to raising steam quickly or sudden cooling, is overcome. Each section is made up of a series of straight tubes expanded at their ends into sinuous steel boxes known as "headers", exactly as in the land type already described.

Drum.—Extending across the front of the boiler, and connected to the upper ends of the front headers by short tubes, is a horizontal steam-and-water drum. As the upper ends of the rear headers are also connected to this drum by horizontal tubes, each section is provided with an inlet and outlet for steam and water.

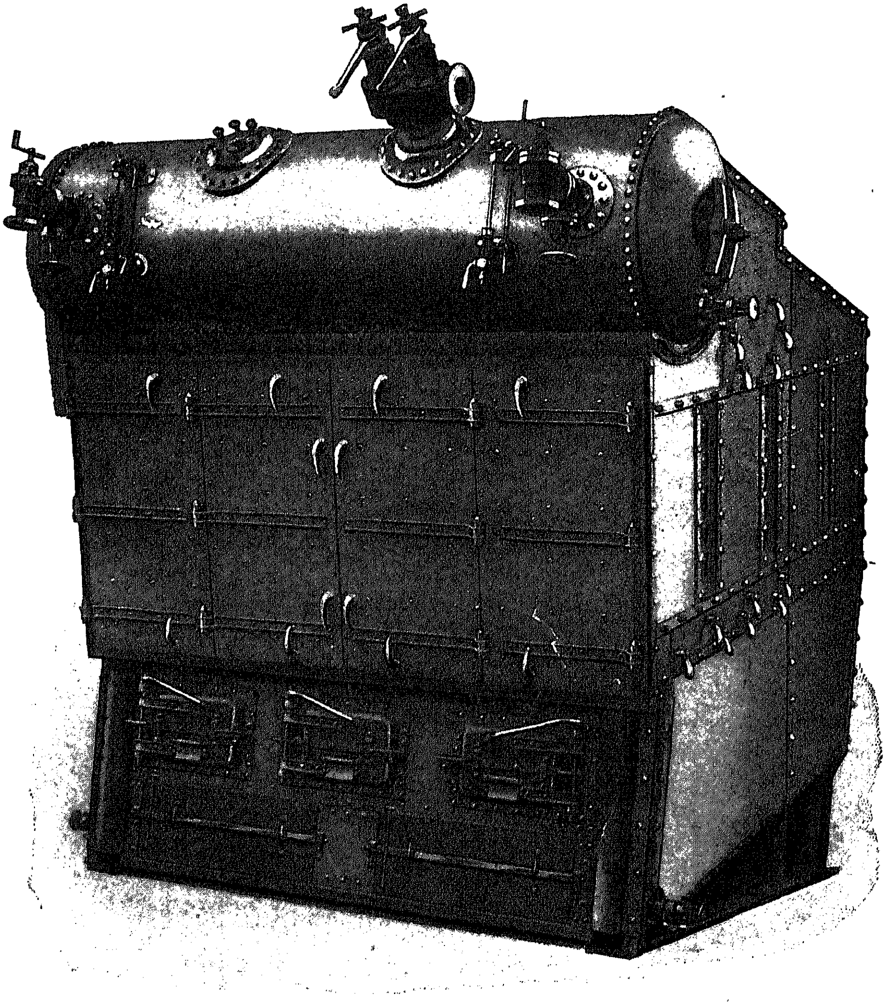


Fig. 1256.—Babcock & Wilcox Water-Tube Boiler—Marine Type

Mud-Drum.—Placed across the bottom of the front headers, and connected thereto by short nipples, is a forged-steel box of square section. This box, being situated at the lowest point of the bank of tubes, forms a receptacle for deposit, and to it the blow-off cock is attached.

Circulation.—The circulation of the water is as follows:—Heat being applied to the inclined tubes, and vapour formed, the mixture of water and steam rises to the high end, and flows through the uptake headers and horizontal return tubes to the steam-and-water drum, the water evaporated in the tubes, and that carried along by the current induced by the steam bubbles, being replaced by water flowing directly from the bottom of the drum downwards through the front headers.

Upon entering the drum the steam and circulating water are directed against baffle-plates, which cause the water to be thrown downwards, while the steam separates and passes round the ends of the baffle-plates to the steam space, from which it is taken by a perforated dry pipe to the stop-valve.

There is thus a continuous circulation of water in one direction, not hindered by any counter current, and this continuous circulation gives the boiler an equal temperature in all its parts, so that undue strains are avoided.

Handholes.—Opposite the end of each tube is an opening or hand-hole in the header, through which the tube can be examined, cleaned, or renewed, each opening being closed by a forged-steel door and stud, the door being drawn up to its seat by means of a forged-steel cap and nut. In the case of the large tubes the joint is made on the inside of the header by an asbestos wire-woven gasket; the openings for medium-size tubes are closed by plugs and coned joints drawn up to their seats by faced cap-nuts, thus making metal-to-metal joints.

Furnace.—The furnace is either built of ordinary fire-bricks or of light fire-tiles, which by a special arrangement are bolted to the side plates. The whole is encased in a special arrangement of plating fitted with non-conducting material, which is so effective in preventing the radiation of the heat that the outside of the casing is quite cool.

This type of boiler has been installed in several electric-light stations, notably in those of the Metropolitan Boroughs of Shoreditch and St. Pancras, in London, the Burgh of Partick, near Glasgow, and the Corporation of Glasgow.

Below will be found extracts from a report of Admiral George Melville, Engineer-in-Chief to the United States Navy, which will prove interesting.

"The task I have set myself to-day is no mean one. I desire to show that the decision to use nothing but water-tube boilers in our future war vessels is a step in advance, and that it is a natural step towards the evolution of the perfect fighting machine. I desire to show that it is no radical change, and that it does not involve the use of anything but a tried, successful, and reliable apparatus that gives us positive and great advantages over the character of boilers heretofore generally used. I desire not to minimize the disadvantages following this change, but to show that these disadvantages are not only not insurmountable, but, for war ships, they have already been overcome.

"Water-tube boilers have advantages and I have never been blind to them. Two years ago I stated that their disadvantages had been sufficiently removed to justify their use on our warships.

"The following table shows the relative economy of cylindrical and water-tube boilers:—

	Babcock & Wilcox.		Single-ended Cylindrical.			
	Annapolis.	Marietta.	Newport.	Princeton.	Vicksburg.	Wheeling.
Number of boilers	2	2	2	2	2	2
Displacement, tons	1000	1000	1000	1000	1000	1000
Knots per ton of coal at most economical speed	26.38	22.27	18.0	19.6	21.25	16.6
Number of screws	1	2	1	1	1	2
Grate surface, square feet ...	98	94	78	78	78	60
Heating surface, square feet	3620	3664	2524	2524	2524	2508

"The increased grate surface we have required with water-tube boilers will be a positive advantage to our ships' steaming qualities. I consider that sustained sea speed depends largely upon the grate surface. Heating surface, of course, must be provided, but I should prefer an excess of grate surface to an exceedingly high rate of heating surface to grate.

"Up to this time we have had no trouble from salt water or grease in water-tube boilers. Indeed, we could hardly be more troubled by salt water with this type of boiler than we have been with cylindrical boilers. We suffered severely in our short war with Spain from dropped furnaces in cylindrical boilers. I do not think that a properly designed water-tube boiler will give more trouble from the use of impure water, such as sometimes we have at sea, than any other boiler. I do not think these tubes more liable than furnaces to fail from a deposit of scale.

"The fact that water-tube boilers raise steam quickly is of the greatest advantage. I have stated elsewhere that I consider the battle of Santiago to have developed the necessity of the use of water-tube boilers whether it taught us anything else or not, It would have been of the greatest advantage to have had, during the blockade of Santiago, boilers capable of raising steam in less than half an hour. Coal need not have been used to keep all the boilers under steam all the time."

Richardsons, Westgarth, & Co.'s "Nesdrum" Water-Tube Boiler.—This boiler consists of a series of nests of tubes with cylindrical drums at the ends. These drums are connected together by expanded nipple tubes, both in the steam and water space.

The drums or headers are cylindrical, and their ends are spherical, and there are no stayed surfaces. It is claimed that the entire disposition of the heating surfaces tends to economy. Each nest of tubes, with its bottom header acting as a water drum, is a separate boiler itself. The front tubes, being hottest, act as ascending columns for the mixed steam and water, and the back tubes are the downcomers. It is claimed that the vertical nests at the back, being heated by the cooler gases, form an efficient economizer, feed-water heater and purifier. The feed water is discharged into the top drum, and in its passage down the tubes is sufficiently heated to deposit the bulk of the scale-forming matter as mud, which, owing to the vertical position of the tube, falls to the bottom of the drum and is blown out periodically.

The tubes, being inclined at a steep angle, present a most favourable surface to the hot gases; the course of the same upwards is direct and unimpeded, and the water circulation is perfect.

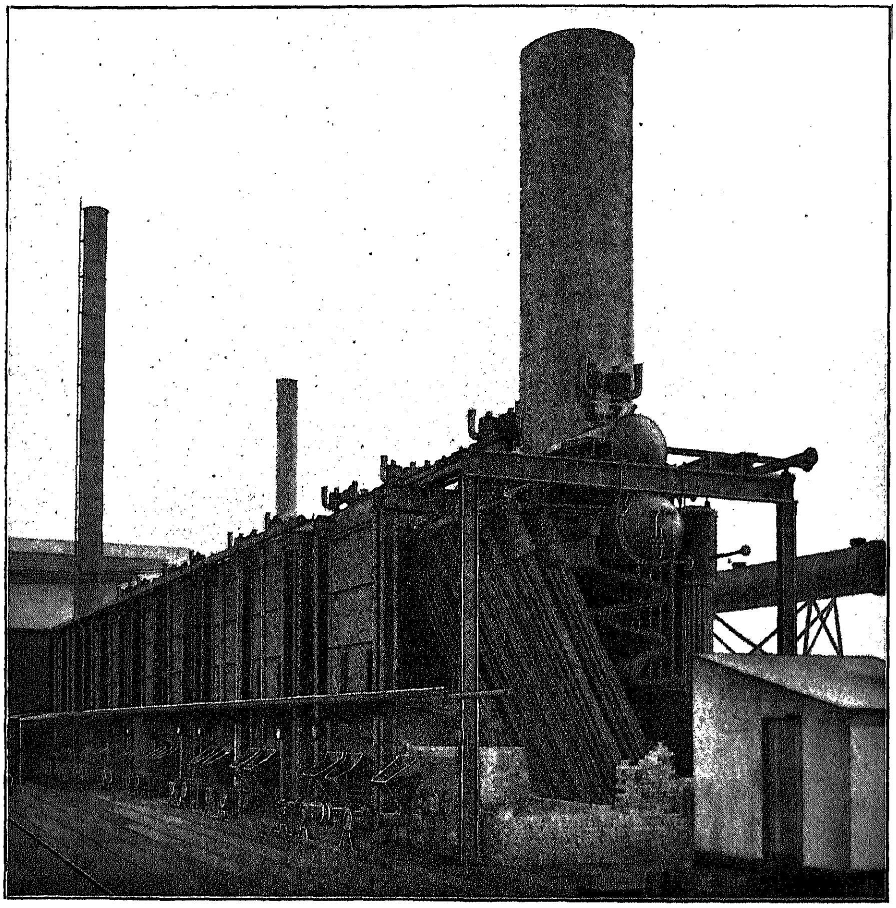


Fig. 1257.—Installation of Richardsons, Westgarth, & Co.'s "Nesdrum" Boilers

The battery of 10 of Richardsons, Westgarth, & Co.'s 555 horse-power "Nesdrum" boilers, as in fig. 1257, is fitted with superheaters and mechanical stokers.

A 48-inch diaphragm steam receiver drum is placed above the ordinary steam and water drum, as this battery of boilers has to cope with very large and sudden demands for steam.

The Stirling Water-Tube Boiler.—This boiler presents the outward appearance shown on fig. 1259–60. Owing to the large ratio of heating surface to grate surface, the long travel of the gases, and the arrangement of the feed, it is claimed for this boiler that the chimney temperature is reduced to the lowest point consistent with a fair chimney draught. Again, as the resistance to the passage of gases is exceptionally small, it is possible to burn a heavy fire on the grate with a very moderate draught in the flues. The back section of the boiler, being isolated from the general circulation, serves as an economizer, and also acts as a purifier, scale and sediment being deposited at this stage, which is farthest from the fire. The feed water enters the back drum at the top of the boiler and descends to the bottom drum where the sediment is deposited, a blow-off cock being provided to allow of its removal.

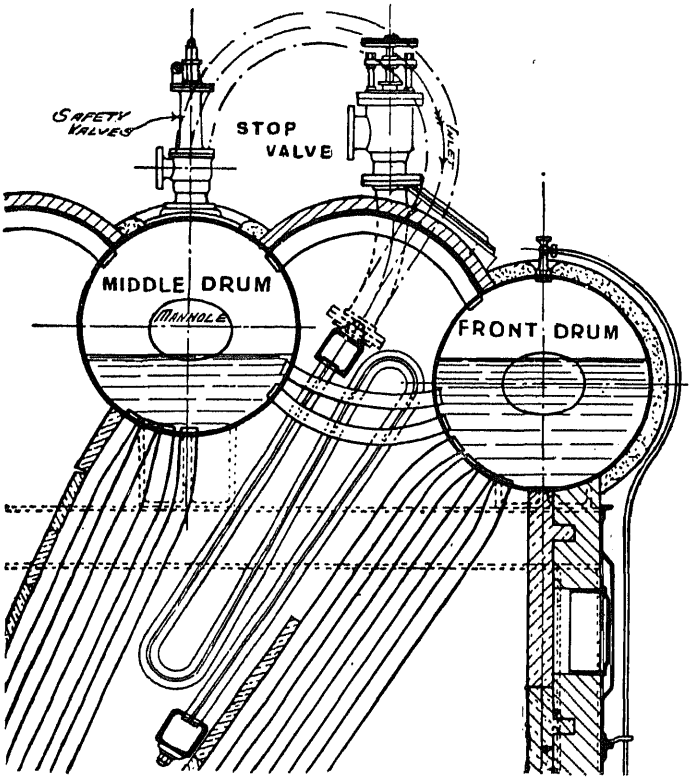
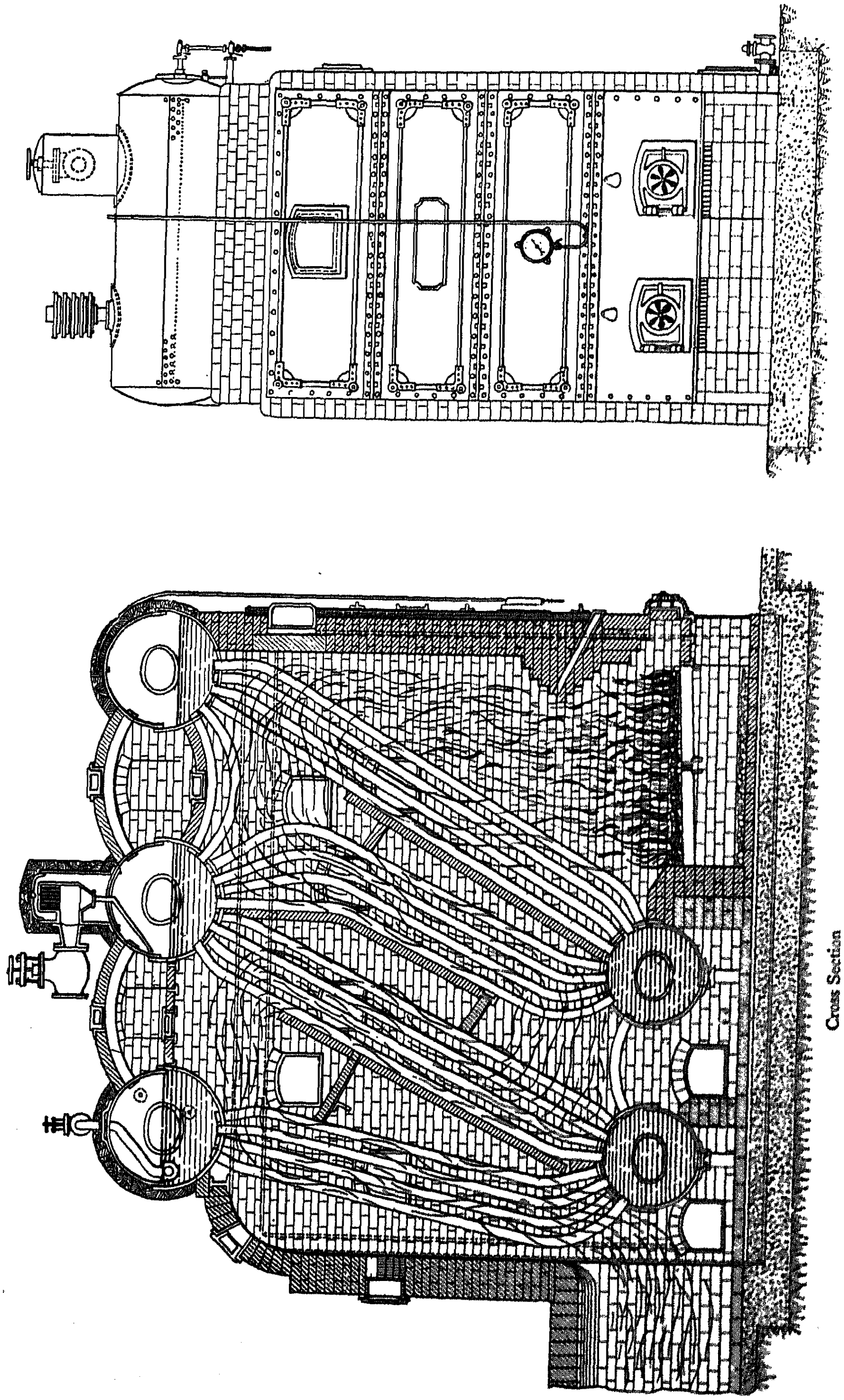


Fig. 1258.—Integral Superheater fitted to a Stirling Boiler

Circulation.—The circulation is rapid and continuous, and the water surface is large for the discharge of steam.

Drums.—Reference to fig. 1259–60, part of which is a sectional elevation, shows that the Stirling boiler consists essentially of three cylindrical steel drums at the upper part and two water-drums below, the latter placed on a level with the fire and behind the fire-bridge. These drums range from about 30 to 36 inches in diameter, according to the size of the boiler, and are connected to one another by a number of tubes not less than



Front Elevation

Cross Section

Fig. 1259-60.—Stirling Boiler Five-Drum Type

3¼ inches in external diameter. The tubes are shaped as shown, in order to permit of their entering the drums at the correct angles.

Baffles.—Brick baffles are arranged between each set of tubes so as to cause the flames and heated gases to be brought into contact with the whole of the heating surfaces of the tubes. The tubes are expanded directly into the drums, and there are no flat surface or stays used. By arranging for the feed-water to enter the boiler at a point most remote from the fire, it is claimed that it is evenly distributed over the back section.

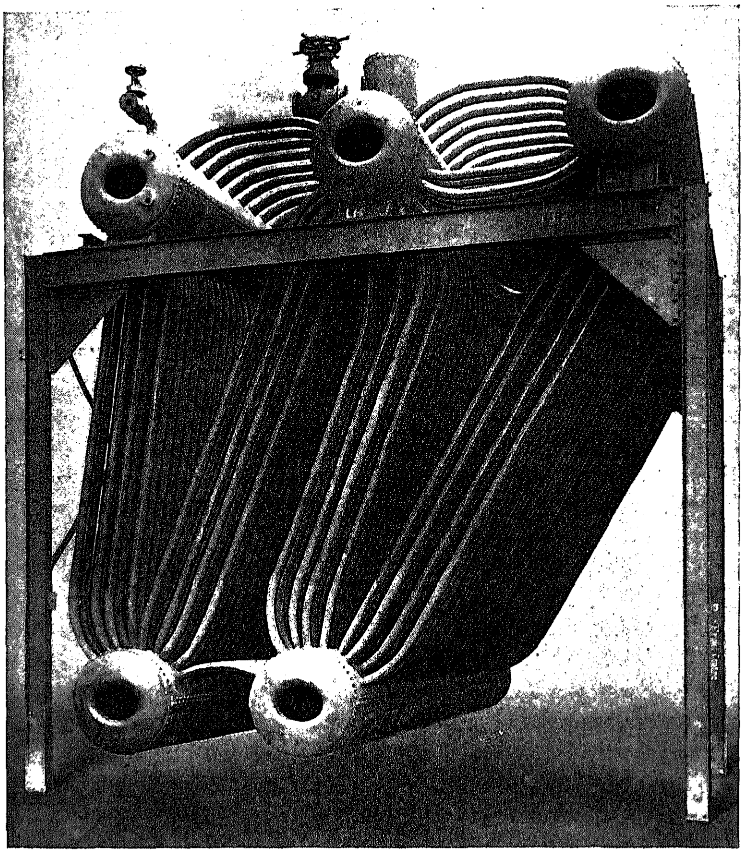


Fig. 1261.—Stirling Boiler, showing Method of Suspension

Mud-Drum.—A mud-drum is provided where the impurities can be blown off as required. It is claimed that the tubes being almost vertical they are not liable to acquire a deposit of dust.

CHAPTER IV

CHOICE OF BOILER

Having shortly described the leading types, the principal considerations to be borne in mind in the choice of a boiler will be discussed. During the last twenty years the rapid progress of electrical undertakings has done a great deal for the improvement of the steam-engine, both in mechanical efficiency and steam consumption; and it may also be said that the introduction of the water-tube boiler, and the consequent competition raised, has had a salutary effect upon the manufacture and design of boilers of all descriptions. In the days of low-pressure and egg-ended boilers, the ability to raise steam in sufficient quantities to enable the engine to turn was all that was aimed at; now everyone is striving to obtain the greatest amount of power from a pound of coal, and while efficiencies as low as 30 per cent are found with vertical boilers, efficiencies as high as 81 or 82 per cent have been reached with water-tube boilers and dry-back marine boilers working with economizers.

In choosing a boiler, the main considerations, stated briefly, are: (1) The amount of water to be evaporated; (2) the space available; (3) the working pressure; (4) the character of the fuel; (5) the character of the demand.

Evaporation.—It is usual to state the amount of water to be evaporated in pounds. In general practice it is astonishing, even yet, to find what a hazy idea some steam users have of their own steam requirements; and again, it may be stated that electrical practice has limited and helped to make the results more exact.

It is very common among engineers of good repute to speak of the horse-power of a steam-boiler. It is well known what confusion exists regarding the horse-power of an engine, whether it be "indicated", "brake", or "nominal". In speaking of the horse-power of a boiler the confusion is even worse, because, given a boiler of known efficiency, the "power" of the boiler depends upon the type of engine to which it has to supply steam. Thus, a boiler capable of evaporating 6000 lbs. of water hourly at 160 lbs. per square inch, would supply steam for an engine indicating 500 horse-power if the latter used only 12 lbs. of steam per I.H.P. per hour; whereas if the engine were of a wasteful type, and used 40 lbs. of steam per I.H.P. per hour, the same evaporation would be required to develop only 150 I.H.P. Yet it would be the same boiler, burning the same fuel, under the same conditions, and doing the same duty *per se*. It could not in the one case be a 500 horse-power boiler, and in the other a 150 horse-power boiler.

In consequence, therefore, of the different quantity of steam necessary to produce a horse-power with various engines, there has been great need of an accepted standard by which the amount of boiler-power required to provide steam for a commercial horse-power may be determined. Watt fixed as a standard 1 cubic foot of water evaporated per hour from 212° F. for each horse-power; this was at the time the requirement for the best engine in use. At the present day, however, it is better to drop the term altogether.

To determine the amount of water to be evaporated, then, the type of engine must be known, and a guarantee or opinion obtained from the makers as to the consumption of steam.

It may be stated briefly, in terms of indicated horse-power per hour, that a—

Single-cylinder non-condensing engine will use ...	30 lbs.
Compound-cylinder non-condensing engine will use	25 „
Compound-cylinder condensing engine will use	18 to 20 „
Triple-expansion condensing engine will use	12 „ 15 „
Large-sized steam turbines will use ...	11 „ 13 „

It is not, therefore, a difficult matter, taking the electrical output in kilowatts and knowing the combined efficiency of engine and dynamo, to arrive at the probable steam consumption of the engine. For example, assume that a boiler is required to supply steam for a 250-kilowatt set having, say, a combined efficiency of 80 per cent. The engine would indicate about 420 horse-power; and suppose the particular engine was of the compound condensing type, using 18 lbs. of steam per I.H.P. per hour, 7560 lbs. of steam would be required hourly. It is necessary to allow some margin beyond this for steam to pumps and auxiliaries.

If, however, the engine were a triple condensing one, using, say, 13 lbs. of steam, and with a combined efficiency of, say, 82 per cent, then the engine would indicate 408 horse-power, and 408 × 13 lbs. = 5304 lbs. of water per hour. With the accurate figures that can be guaranteed by electrical manufacturers, derived from a large experience and exhaustive tests, it is a comparatively easy matter to arrive at the correct evaporation required of the boilers. It is always well to err on the side of being over-boilered, as in an electric station or other works nothing is so troublesome or annoying as being short of steam.

Space.—Regarding the space available little may be said. Knowing the required evaporation, the size of boiler can be properly chosen, and if the engineer decides upon a Lancashire, Galloway, or dry-back, the space occupied by either of these types, including the brickwork setting, is a fixed quantity, and the space must be suitable for these types of boilers. A water-tube boiler, however, is much more flexible, and a greater amount of boiler-power can be got into the same space, or conversely, the same boiler-power can be put into less space, not only saving the cost of land, but also buildings. This will be apparent by reference to the table quoted on p. 18, where a water-tube boiler, with ground at £2 per foot, entails a cost for buildings and ground of £360; while for a Lancashire boiler with economizer the figure is £820; a locomotive boiler, £300. This is for a single boiler, but with a large number of big units, which is only possible where the water-tube type are installed, the figures are even more in favour of the water-tube boiler. Supposing, for instance, an evaporation of 75,000 lbs. hourly is required. Allowing for spares, this would require at least twelve Lancashire boilers 30 feet long × 8 feet in diameter. If these were set in

621·3
N09·5
3198

two batteries of six each, and allowing for flues and stoking space, a boiler-house about 150 feet long × 46 feet wide would be required at the very least. For the same evaporation, and allowing the same proportion of spares, five large water-tube boilers would be required, and could be got easily into a space 80 × 44 feet. The economy of space thus shown is one of the factors which has led to so large an adoption of water-tube boilers in electrical power stations in this country.

High Pressure.—The adoption of triple-expansion engines has led to the use of pressures from 160 to 200 lbs. per square inch, and more; and although shell boilers are made every day for these pressures, it is generally admitted that the small diameters and sectional construction of some water-tube boilers, and the consequent high factor of safety, render them more suitable and less liable to injury than shell boilers, the construction of which for high pressures necessitates abnormally thick plates and special construction. In this connection it may be of interest to quote a passage from the report of the chief engineer of the Engine, Boiler, and Employers' Liability Insurance Co., Ltd., Mr. Michael Longridge, M.A., M.I.C.E., M.I.M.E., on Lancashire boilers for high pressures:—

"The furnaces of the large high-pressure Lancashire boilers now being used are more liable to overheating, and consequent wear and tear, than those of the smaller boilers working with lower pressures which held the field some years ago. For this there are several reasons: the thicker plates required to make flues of larger diameter fit for high pressures; the increased stiffness of the thicker end-plates; the higher rate of combustion possible in the larger furnaces; and the increased evaporation and deposition of scale per square foot of heating surface. All these render more care necessary in selecting pure feed-water and in keeping the plates clean. Indeed, the writer's experience is leading him to believe that the system of staying the front end-plates will have to be modified, or the feed-water will have to be purified, if Lancashire boilers are to keep the place they now occupy in the estimation of steam users."

Fuel.—The character of the fuel is another important point to be considered when choosing a boiler. In electrical stations at present, coal, coke, and town refuse are the fuels used. It is the object of managers, when using coal, to keep their fuel bill as low as possible; whereas, with town refuse, the main object is to destroy it, using the heat generated for some useful purpose. It is possible that in the future boilers will be fired by gas, and if a regular supply of liquid fuel can be obtained it may largely replace coal in the generation of steam.

In a Lancashire boiler, the distance between the bridge and the crown-plate, the form of fire-bars, the air-space between the fire-bars, the form of fire-doors, all constitute important factors in burning fuel to the best advantage. With a low rate of combustion, and helped by an economizer, a very high efficiency may be obtained by a Lancashire boiler. With a high rate of combustion the same efficiency cannot be maintained, as the space for combustion is not only confined, but the fuel and gases are surrounded by cool boiler surface. On the other hand, with a water-tube boiler having a fire-brick furnace, a high efficiency with a high rate of combustion may be obtained, and the form of the fire-brick furnace can be varied to suit the kind of fuel to be used. By the help of mechanical stokers much labour may be saved, smoke prevented, and a cheap low-grade fuel used.

If coke is the fuel, the form of grate-bar must be very carefully considered. The heat from coke is local, and unless precautions are taken, much clinker is formed, and the fire-bars suffer severely; and either water-pans in the ash-pit, or steam-jets, or both, are used to minimize this.

Character of Load.—The character of the demand for steam is another factor which must be determined in the choice of a boiler. In an electric-lighting station the load varies very considerably, and in case of fog a sudden demand is made for steam.

In a tramway or electric-railway system the load, while showing very rapid fluctuations over short periods of time, maintains a fair average during the day if the service is regular; but at times, when people are going to and returning from work, or on a general holiday, an extra demand is made upon the power station. A boiler which can raise steam rapidly, economically, and without injury to itself, should therefore be chosen. In a mill, where there is a steady load and no possibility of a sudden demand, this consideration is not of such vital importance; but still the possibility of raising steam rapidly can never be considered a disadvantage.

CHAPTER V

THE STRENGTH OF MATERIALS

Material.—The material used for making boilers at the present time is almost invariably mild steel. It can be rolled, hammered, flanged, and welded. It is very liable to be injured when overheated, and great care has to be taken when heating in the furnace. Careful annealing should always be resorted to. The tensile strength of steel plates used for boiler-making should be from 26 to 30 tons per square inch, with an elongation of 20 per cent in 10 inches.

Test-Strips.—It is customary to cut the test-strip about 20 inches long, as this is the usual length to go into the testing-machine. Two marks are punched on the strip exactly 10 inches apart, from which the percentage of elongation is measured after the strip has been pulled in the machine. For furnaces where the plates are exposed to the impact of the flame, the tensile strength should be between the limits of 24 and 28 tons per square inch, with an elongation of 25 per cent in 10 inches. Plates should stand bending to a radius of one and a half times the thickness of the plate, and until the sides are parallel at a distance of not more than three times the thickness of the plate. The test-pieces should be about 2 inches broad, and carefully prepared.

Annealing.—A cracked plate is generally due to the fact that it has been heated locally. The crack takes place sometimes immediately, sometimes months after the boiler has been set to work. Steel plates should, therefore, after being heated locally, be heated all over to a dull red, and allowed to cool gradually. All flanging, so far as practicable, should be done in one operation; hydraulic appliances are used for this purpose. In making boilers of the shell type, the large diameters do not allow of this, and consequently plates after being finished have to be heated all over to a bright red, and then allowed to cool gradually. If a plate will not bend when cold the material is not fit for use, and should be rejected.

Drilled Holes.—Until comparatively recent times it was customary to punch the rivet holes in boiler plates; but this process impairs the strength of the material, and tends to make it unreliable. In modern boiler works suitable multiple drilling-machines are found for drilling the boiler shells after they have been bent to the proper curvature.

Rivet Bars.—The mean tensile strength of rivet bars should not be less than 26 tons per square inch, nor more than 30 tons per square inch, with an elongation of about 25 per cent in 10 inches. The rivets should be made from ingot steel bars. Rivets have a slightly higher tensile strength than the bars from which they are made. We give here model specifications for tubes, plates, and rivets.

SPECIFICATION FOR BOILER SHELL-PLATES

Material. The plates to be of Siemens-Martin open-hearth mild steel.

Tests. The tensile strength to be from 26 to 30 tons per square inch, with an elongation of not less than 20 per cent in 10 inches.

Quenching and Bending Tests. A strip 2 inches broad and 10 inches long, sheared from the plate, and heated to redness, then cooled in water of 80° F., must stand bending until the inside radius of the curve is one and a half times the thickness of the plate.

The plates must stand this bending without showing any signs of fracture.

Thickness and Weight. The plates are to be as near as possible of the specified thickness and weight, with the usual allowance.

Finish. The plates to be free from buckles, surface defects, hard spots, and laminations.

Test-Pieces. Two pieces are to be cut from each plate; one is to be tested by the maker, the results of such test being sent the purchasers with each delivery, and the second test-piece is to be retained by the maker for six months against further requirements.

Maker's Brand. All plates are to be branded with the maker's stamp, the cast number, and marks as may be specified.

Weights. The weight is to be painted on each plate in good permanent white paint.

SPECIFICATION FOR FURNACE PLATES

Material. The plates to be of Siemens-Martin open-hearth mild steel, welding and flanging quality.

Tests. Tensile strength to be from 24 to 28 tons per square inch. Elongation not less than 25 per cent in 8 inches, preferably 27 per cent.

Quenching and Bending Tests. A strip 2 inches broad and 10 inches long, cut from the plate, must endure bending double, and closing flat, both before and after being heated to flanging heat, without showing any signs of fracture.

Thickness and Weight. The plates to be as near as possible of the specified thickness and weight, with the usual allowance.

Finish. The plates to be free from buckles, surface defects, hard spots, and laminations.

Test-Pieces. Two test-pieces are to be cut from each plate; one is to be tested by the manufacturer, the result of such test being sent to us with each delivery on the usual form, and the second test-piece is to be retained by the manufacturer for six months against further requirements.

Maker's Brand. All plates are to be branded with the maker's stamp and cast number.

Specification for Lap-Welded Steel Boiler-Tubes

Material. The steel to be made by the open-hearth process.

Workmanship. The tubes are to be lap-welded, and to be perfectly straight, smooth, cylindrical, of uniform thickness, and of equal diameter throughout, excepting where specified of increased diameter at the ends. They are to be free from scale, seams, grooves, or blisters, either internally or externally.

No patched or brazed tubes will be accepted.

Ends. The ends of the tubes, when required, are to be swelled to the gauges supplied by the purchaser. All ends to be carefully annealed.

Tests. Pieces taken from the strips of which the tubes are made, or from the tubes themselves, shall stand the following tests:—

Ultimate tensile strength—22 to 25 tons.
Elongation—Not less than 25 per cent in 8 inches.

The tubes shall stand expanding at the ends cold, to an increase of 5 per cent in diameter, by means of an ordinary boiler-tube expander, without splitting, cracking, or opening the weld.

Strips 1½ inch wide, cut lengthwise, flattened out, heated to a low cherry-red, and cooled in water, shall stand bending double over a radius of ½ inch without fracture.

All tubes to be tested, before leaving the maker's works, to 1000 lbs. hydraulic pressure, and certificates of the test to be provided.

Specification for Weldless Steel Boiler-Tubes

Material. The tubes are to be made of Siemens-Martin, acid open-hearth steel, of the best quality.

Workmanship. The tubes are to be weldless hot-rolled, and to be well finished, and clean inside and outside, and free from seams, blisters, grooves, and laminations.

They are to have an equal sectional thickness all round, as nearly as possible, but in no case must the thickness, at any part, be less than the specified thickness; neither must it exceed the specified thickness at any part by more than 1½ S.W.G.

All the ends are to be annealed, and clean, and smooth.

Tests. Pieces taken from the strips of which the tubes are made, or from the tubes themselves, shall stand the following tests:—

Ultimate tensile strength—21 to 25 tons per square inch.
Elongation—25 per cent in 8 inches.

The tubes are to stand expanding at the ends cold, to an increase of 10 per cent in diameter, by means of an ordinary boiler-tube expander, without splitting, cracking, or showing any signs of fracture.

Strips $1\frac{1}{2}$ inch wide, cut lengthwise or crosswise, flattened out, heated to a low cherry-red, and cooled in water, must stand bending over a radius of $\frac{1}{2}$ inch without showing any signs of fracture.

All tubes to be tested, before leaving the maker's works, to [illegible] lbs. per square inch, hydraulic pressure, and a certificate of the test is to be supplied with every shipment.

SPECIFICATION FOR STEEL RIVETS

Material.

The rivets are to be made from Siemens-Martin open hearth acid steel, having a tensile strength of not less than 24 tons and not more than 27 tons per square inch, with a minimum elongation of 25 per cent in 8 inches.

Tests.

Pieces cut from the bars, heated uniformly to a low cherry-red, and cooled in water of about 80° F., must stand bending double in a press to a curve of which the inner diameter is equal to the diameter of the bar tested. Samples from each batch of rivets are to stand the following forge tests satisfactorily without fracture:—

(*a*) The rivet to be bent double, cold, to a curve of which the inner diameter is equal to the diameter of the rivet tested.

(*b*) The rivet to be bent double when hot, and hammered until the two parts of the shank touch.

(*c*) The head to be flattened when hot, without cracking at the edges, until its diameter is two and a half times the diameter of the shank.

(*d*) The rivet, when heated to a dull red, to be compressed to a third of its length, after which it is to be punched through without showing any signs of fracture or cracks.

(*e*) The shank of a rivet is to be nicked on one side, and bent cold, to show the quality of the material.

Finish.

The rivets to have the heads truly central, and the shanks practically parallel.

Certificate of Tests.

A certificate to be supplied, stating that the above tests have been carried out, and are in accordance with this specification.

CHAPTER VI

MOUNTINGS

As the safe and easy working of a boiler largely depends on the mountings, special consideration should be given to them.

Safety-Valves.—There are three leading types of these—the dead-weight, the ball-and-lever, and the spring safety-valve. The ball-and-lever valve is very little used in electrical stations, and will therefore not be described.

The dead-weight type (fig. 1262) consists of a valve directly loaded by cast-iron plates, which are lifted when the internal pressure on the valve becomes great enough to overcome their weight, the steam escaping between the valve and its seat by way of openings in the valve-cover to the atmosphere. The advantage of this type is that it is not easy for an attendant to tamper with it, as a very heavy weight must be added to cause any appreciable rise in the pressure at which it blows off.

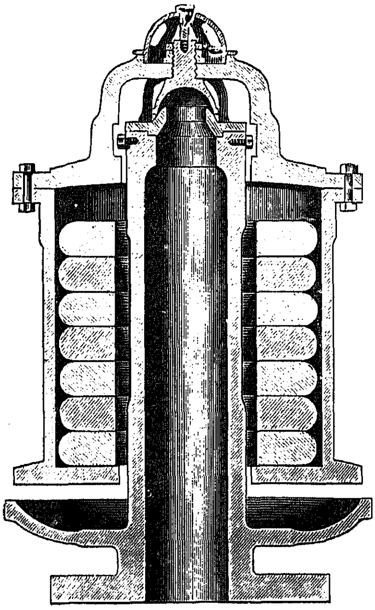


Fig. 1262.—Dead-Weight Safety-Valve

Messrs. Hopkinsons of Huddersfield make a safety-valve (fig. 1263) which lifts and blows off steam whether the pressure be too high or the water become too low. It consists of a valve loaded externally by means of dead-weights, or ball and lever; through the centre of this valve there is an orifice, in which is the seat of an auxiliary valve. A spindle is attached to the auxiliary valve and descends into the boiler, a weight being suspended at its lower end, keeping the smaller valve on its seat; and attached to the boiler shell, by means of an eye-bolt, is a double lever, at one end of which is a counter-weight and at the other end a float. Should the water in the boiler become low it leaves the float, and the latter falls. The spindle works through an eye in the double lever; on the spindle immediately above the double lever is an adjustable collar, so that when the lever rises as the water falls, it comes in contact with the adjustable collar, and

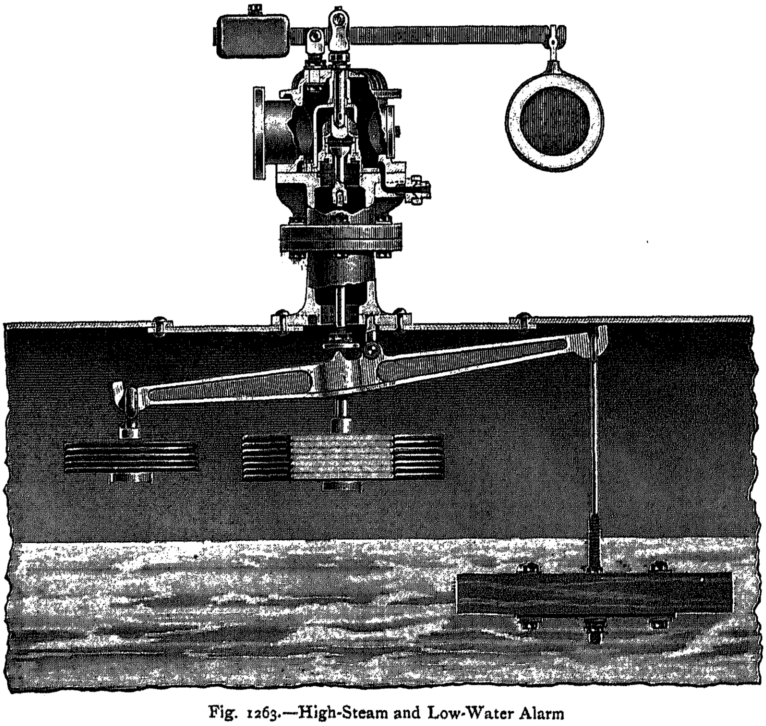


Fig. 1263.—High-Steam and Low-Water Alarm

this lifts the secondary valve above off its seat, allowing the steam to escape.

In spring safety-valves the place of the dead-weight is taken by a spring; it should be so designed that should the spring break, the valve cannot fly off, and that the compression of the spring cannot be increased by the attendant (fig. 1264).

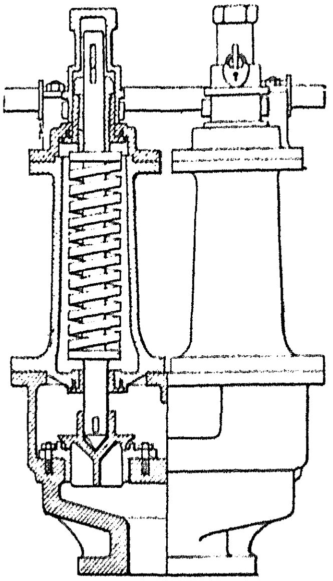


Fig. 1264.—Spring-Loaded Safety Valve

Size of Safety-Valve.—In calculating the size of a safety-valve the following points should be taken into account:—

1. The grate area, because the larger the grate the greater the amount of coal burned, and therefore more steam produced.
2. The heating surface of the boiler taken in conjunction with the type of the latter.
3. The pressure of steam. The volume of steam varies approximately inversely with the pressure, and consequently for high pressures the area of the safety-valve may be smaller than for low pressures, while the velocity at which the steam escapes will be very much higher.

Below are given the Board of Trade proportions for marine boilers for the area of safety-valves, from which it will be seen that the area is considered in relation to the grate surface and the pressure only irrespective of draught, rate of combustion, and ratio of grate surface to heating surface.

BOARD OF TRADE REGULATIONS REGARDING SAFETY-VALVES

Boiler Pressure in lbs. per square inch.	Area of Safety-Valve in square inches per square foot of Grate Surface.
50	0.576
60	0.500
70	0.441
80	0.394
90	0.357
100	0.326
110	0.300
120	0.277
130	0.258
140	0.241
150	0.227
160	0.214
170	0.202
180	0.192
190	0.182
200	0.174

Low-Water Alarm.—A simple arrangement for giving the attendant notice when the water is too low is an apparatus in which a whistle is blown before the water has disappeared from the gauge-glass. The whistle is blown by steam issuing from a valve which is opened by a float somewhat similar to the Hopkinsons' alarm already described (fig. 1265).

Fusible Plug.—In locomotive and shell boilers this is placed in the crown of the furnace; and should the furnace plates become bare of water and overheated, the fusible metal forming the plug melts, and steam is discharged into the furnace, putting out the fire.

In cleaning boilers care should be taken that no scale is allowed to accumulate over the plug, as, if this happens, the fusible metal itself becomes overheated although there may be plenty of water in the boiler, the plug drops out, and steam and water are blown on to the fire when it is least expected.

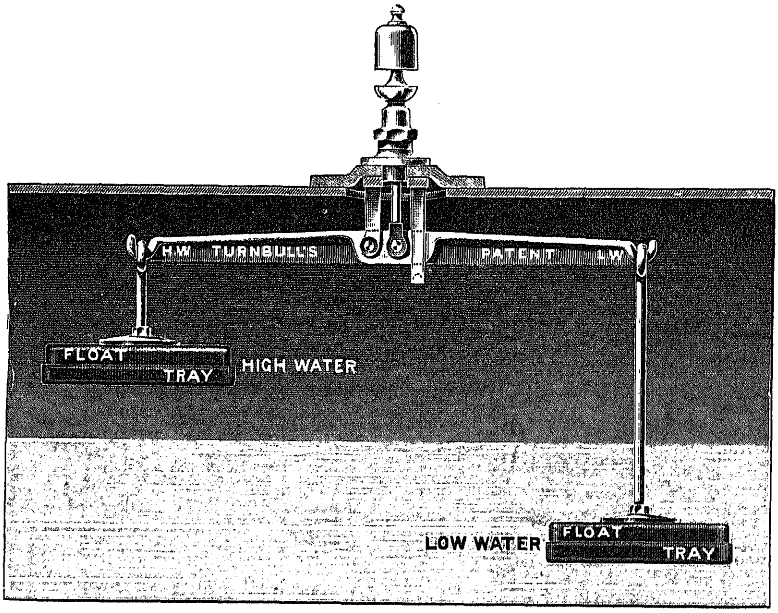


Fig. 1265.—Low-Water Alarm

Water-Gauges.—The level of the water in the boiler is indicated by a water-gauge. This consists of a glass tube connected to the boiler by two arms, as shown in fig. 1266—one connected with steam-space, and the other with the water-space at about equal distances from the proper working level of the water in the boiler. In each of these arms is a cock, so that both the water and steam can be shut off while putting in a new glass. The ends of the glass are connected to the arms through stuffing-boxes, which should be of such a construction that a glass can be taken out and replaced by a new one as quickly as possible. It is most important that the true level of the water in the boiler should be indicated by the water-gauge, and this can only be assured when all passages are kept clear. For this purpose a drain-cock is attached to the bottom arm, by which the passages in the gauge can be blown through and tested. This blowing through is not done properly by simply opening and closing the drain-cock, as this does not give a sufficient scour, nor does it necessarily give any indication if anything is wrong. The water-arm cock should first be closed and the drain opened, in order to scour and test the steam-arm passage and the glass; then the steam-arm cock should be closed and the drain opened to scour and test the water-arm passages. It is also important that cleaning-screws should be fitted to allow a wire being passed through all passages in the gauge right into the boiler, in order to remove any obstruction.

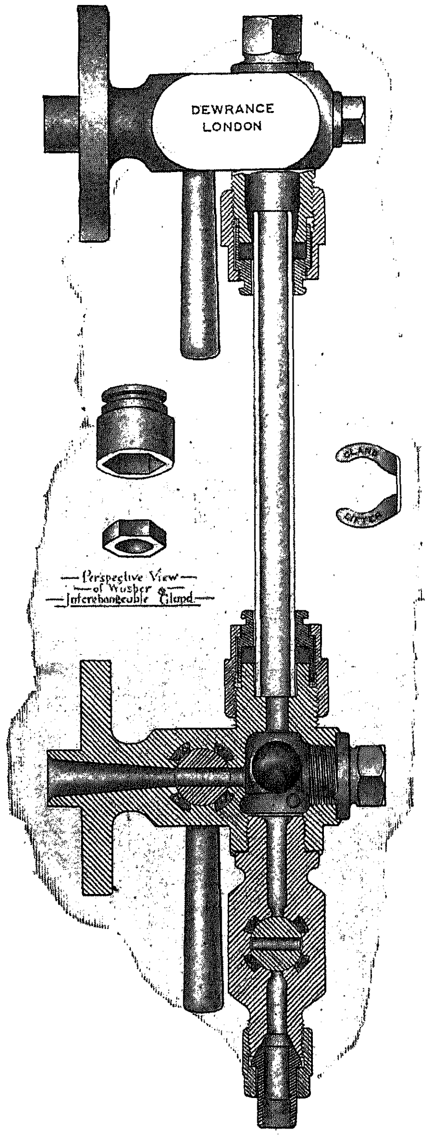


Fig. 1266.—Water-Gauge Fitting

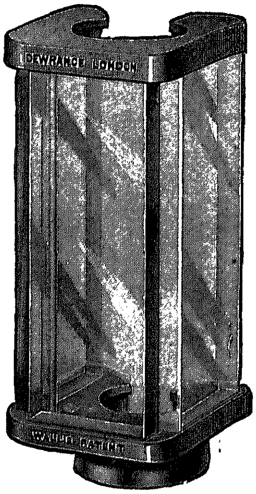


Fig. 1267.—Gauge-Glass Protector

A water-gauge would be a simple apparatus, were it not for the frequent bursting of glass tubes and consequent danger to attendants from broken particles of glass or the outrush of steam and water. The first of these dangers is guarded against by the use of a protector (fig. 1267) fitted to the gauge. This should be made of a toughened glass which will stand the impact of the broken pieces of glass, and arranged so that the water-level in the glass can be easily seen both from the front and sides. The protector should be easily removed and replaced for cleaning and putting in a new glass.

The outrush of scalding water is automatically prevented by a ball which lies in the lower arm, and which comes into action when the glass breaks. The outrush of steam does not constitute such a danger as that of the water, and many engineers do not care to interfere automatically with this, especially as by doing so they add considerable complications to the gauge, owing to the arrangements which must be made to ensure the ball closing the passage when the glass breaks, while leaving it quite clear when the glass and steam-arm passages are being blown through for cleaning. This has been successfully accomplished by various makers. All automatic balls should be accessible for inspection and cleaning without putting the gauge out of work for any length of time.

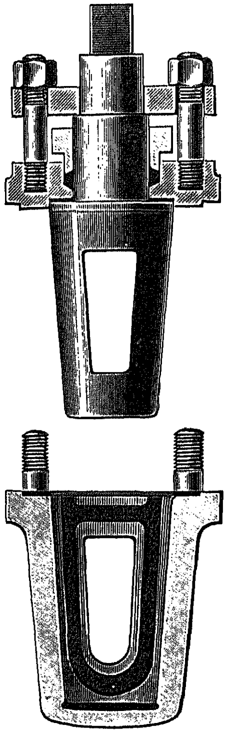


Fig. 1268.—Blow-off Cock

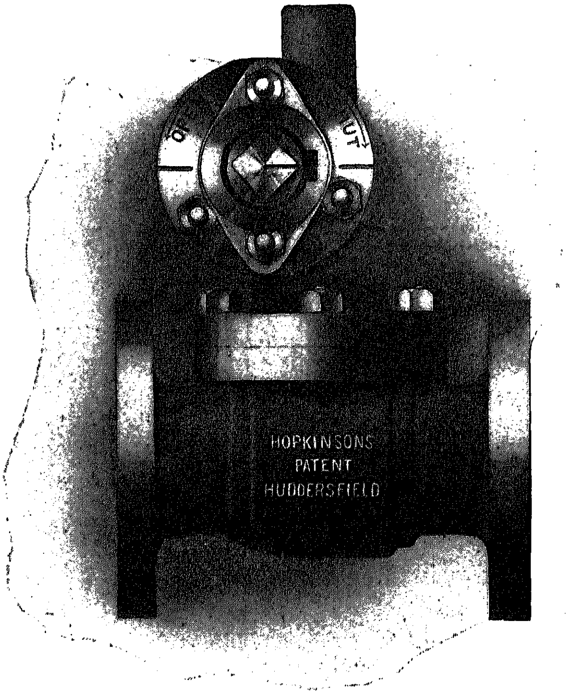


Fig. 1269.—Blow-off Valve, with Locking Gland

This very important blowing through of water-gauges is much simplified in a recently patented gauge, in which the whole operation is performed by working one handle, which actuates a simple three-way cock working through the following cycle:—First, working position; second, blow through water-arm passages with steam passages shut; third, blow through steam passages and glass with water passages shut. Every passage in the gauge is therefore blown through by either steam or water (never both together) by the working of the one handle, which can be brought down to within easy reach of the boiler attendant by a lengthening rod.

Blow-off or Sludge Cock is fitted to the lowest part of the boiler at a position where sludge will collect (fig. 1268). By the opening of this cock the sludge is carried away by the outrush of water. These cocks are generally of 2- to 2½-inch bore, and are made either of the asbestos-packed type or of the full-way valve type.

In the former the plug of the cock works on a surface of asbestos, which is packed into grooves surrounding the water-way. This asbestos is sufficiently elastic to allow the plug to expand without seizing. The plug should be tapered, and have an arrangement for holding it on to its seat, which should be adjustable to take up any wear.

Full-way valves are made with the valves either wedge-shaped or parallel. In the former the wedge forces the valve against its seat, while in the latter a spring is employed.

When valves are used for the blow-off it is desirable that they should open full by the turning of the handle through a quarter of a revolution, so as not to differ from the cock which is now generally used.. This is best accomplished by employing a rack and pinion.

Blow-off cocks and valves are sometimes fitted with a "locking gland" (see fig. 1269), which prevents the key being removed until the cock has been fully closed. This locking gland is sometimes made to fulfil another purpose when there is a range of boilers; by having only one key in the station, which it is impossible to remove from any cock until the cock is fully closed, it is ensured that only one cock in the range can be open at one time, thus preventing the scalding of any boiler-cleaner who chanced to be in an empty boiler with blow-off cock open when an adjacent boiler is being blown off.

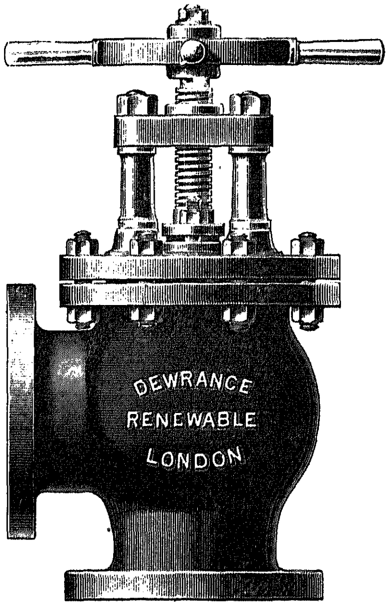


Fig. 1270.—Steam Stop-Valve

Another way of preventing the possibility of an accident of this nature is to fit a back-pressure valve on the outlet side of each blow-off cock.

Blow-offs, whether of the cock or valve type, are often a source of trouble to engineers owing to the cutting action of the sediment blown through it. The cutting action is most severe when the cock or valve is only partially open, and this trouble is obviated by having an auxiliary valve fitted to the outlet side of the blow-off cock. When blowing off, this auxiliary valve is kept closed until the cock has been opened full. The gradual opening for blowing through is then done by the auxiliary valve, which may cut, as it is not necessary that it should be absolutely tight. This auxiliary valve may be made to act as an isolating valve, and then fulfils two functions.

The Steam Stop-Valve, through which all steam to the engines must pass, is usually fixed on the crown of the boiler (fig. 1270). The body is either of cast-iron or steel, and the spindle, valve, seat, and the working parts are of a good gun-metal or bronze. The valve disc is of the mushroom type, with either flat or mitre faces, and has a suitable guide to ensure it coming truly on to its seat. Most makers are now manufacturing the valve seat in such a way that it can be taken out and replaced by a new one. In these valves the seat is not driven into the cast-iron or steel body, but is a loose fit, the joint being made on the horizontal plane between the body and a collar formed on the seat. The pressure required for making this joint is either got by tightening the cover bolts (there being a distance-piece or cage between the cover and the seat) or by some arrangement of set-screws which hold the seat down. The joint being on the horizontal plane the seat is free to expand and contract independently of the body; it is therefore not liable to become distorted as in the old form, in which the joint was made in the vertical plane by the seat being driven tightly into the body.

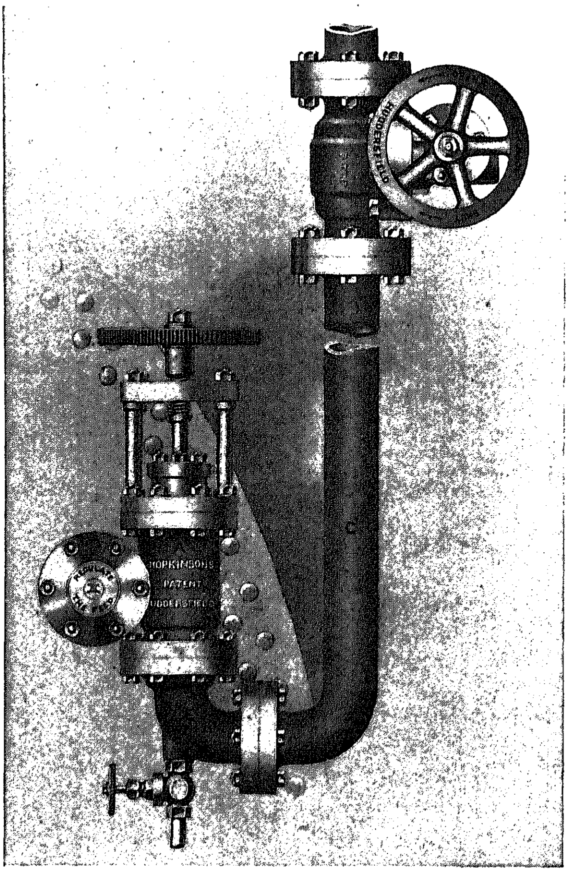


Fig. 1271.—Accessible Feed-Valve

Isolating Valve.—In a range of boilers an isolating or back-pressure valve is sometimes fitted on the outlet side of each stop-valve. These isolating valves act as a safeguard to the boiler-cleaners should anyone open the stop-valve while they are working in any empty boiler. The stop-valve itself is sometimes made to act as an isolating valve by the valve not being attached to the spindle, which allows it to shut automatically should an accident occur. Some makers are manufacturing isolating valves to work on both sides, so that they would also shut automatically in the event of a steam-pipe bursting in the station. These, however, require careful adjustment, in order that they do not come into action by the ordinary flow of steam from the boiler to the engine.

Feed Check-Valves are of somewhat similar construction to the stop-valve already described, except that the valve disc is not attached to the spindle, so that it closes with the back pressure (fig. 1271). By the addition of an asbestos-packed cock or auxiliary valve the check-valve is made accessible for inspection and repairs while the boiler is under steam. With some waters it is advisable to have a cleaning branch, through which the internal feed-pipe may be cleaned. In working, this branch is closed by a blank flange.

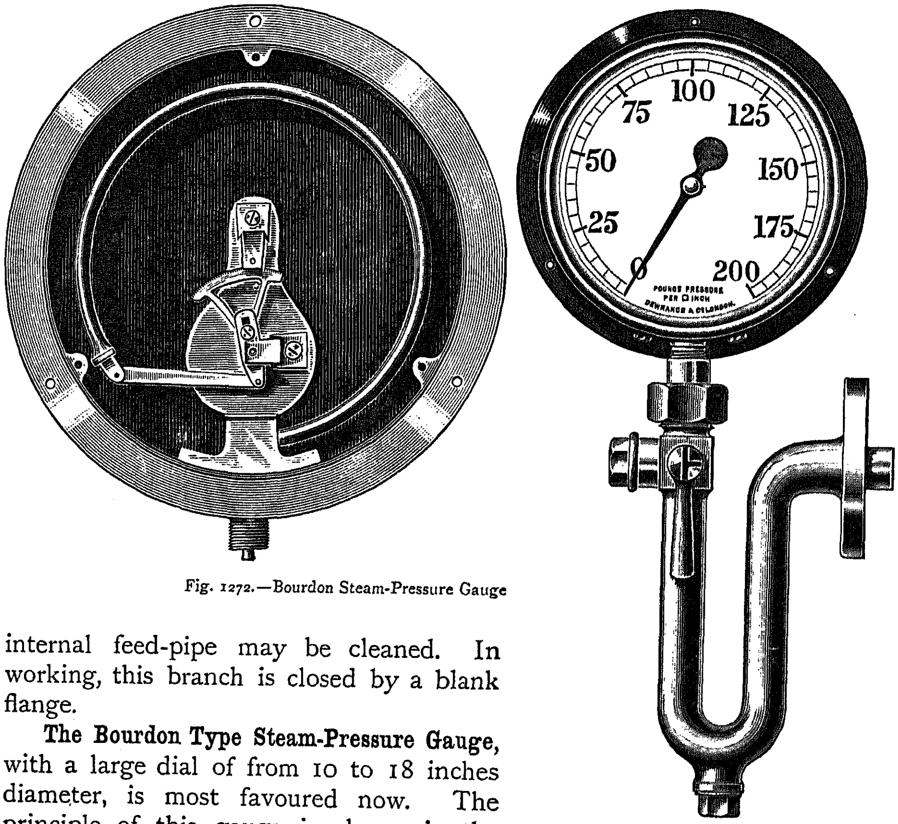


Fig. 1272.—Bourdon Steam-Pressure Gauge

The Bourdon Type Steam-Pressure Gauge, with a large dial of from 10 to 18 inches diameter, is most favoured now. The principle of this gauge is shown in the illustration, fig. 1272, which shows the back of a gauge with cover plate removed. The bronze or steel bent tube, which tends to straighten under internal pressure, has one end fixed, while the other end, which is free to move, is connected to gearing consisting of a quadrant working into a pinion. On the spindle of this pinion is fixed the indicating pointer.

Some consideration should be given to the cock and siphon, by which these gauges are connected to the boiler. It is advisable that the cock should have provision for a connection to a test gauge. The siphon should make an effective water-seal, preventing any steam getting into the gauge, and should be of sufficient size to cool the water contained in it by radiation.

CHAPTER VII

CARE OF BOILERS

Safety-Valves.—Great care should be taken to see that these valves are in working order. Overloading or neglect frequently leads to the most disastrous results. Safety-valves should be tried at least once every day, to see that they act freely.

Pressure-Gauge.—The steam-gauge should stand at zero when the pressure is off, and it should show the same pressure as the safety-valve when that is blowing off. If not, then one is wrong, and the gauge should be tested by one known to be correct.

In boilers standing idle particular care should be exercised to see that water lying in the siphon pipe, or in the gauge, should not freeze in cold weather.

Water-Level.—The first duty of an attendant before starting, or at the beginning of his watch, is to see that the water is at the proper height.

Water-Gauges.—Water-gauges must be kept clean and be blown out frequently, and the glasses and passages to gauge kept clean also. When a boiler has stood unused, before lighting the fire the water-gauge should be tested as to cleanliness and the freedom of the water and steam passage. Screws are provided in the water-gauge cocks, which enable a rod or wire to be put right through the water-way. More accidents are attributable to inattention to water-gauges than to all other causes put together.

Feed-Pump or Injector.—These should be kept in perfect order and be of ample size. No make of pump can be expected to be continuously reliable without regular and careful attention. It is always safer to have two means of feeding a boiler. Check-valves and self-acting feed-valves should be frequently examined and cleaned.

Low Water.—In case of low water, immediately cover the fire with ashes (wet if possible) or any earth that may be at hand. If nothing else is handy, use fresh coal. Draw the fire as soon as it can be done without increasing the heat. Neither turn on the feed, stop engine, nor lift safety-valve until fires are out and the boiler cooled down.

Blisters and Cracks.—These are liable to occur in the best material. When the first indication appears there must be no delay in having it carefully examined and properly cared for.

Firing.—Fire evenly and regularly, a little at a time. Spreading the fire gives the best results. Fires should not be carried too thick. Take care to keep grates evenly covered, and allow no air-holes in the fire. Do not "clean" fires oftener than necessary. Keep the fuel off the dead-plate.

Cleaning.—All heating surfaces must be kept clean outside and in or there will be a serious waste of fuel. The frequency of cleaning will depend on the nature of the fuel and water. With the scale produced from some waters, even a thin crust may lead to overheating. Proper intervals for cleaning should be established by experience. In the case of water-tube boilers it is easy to determine the proper period for cleaning, because it is a simple matter to remove the handhole fittings in front and rear and look through the tube. Any accumulated scale is removed by the scrapers, but care should be observed that the scale pushed to the back end of the tubes should not be allowed to remain there, but should be removed. Water through a hose will facilitate the cleaning, and the best time to clean a boiler is immediately after the water has been removed from it. The fire should be drawn, the brickwork allowed to cool, and the incrustation attacked with a scraper whilst it is in a wet state.

Hot Feed-Water.—The use of hot feed-water is a source of great economy.

Foaming.—When foaming occurs in a boiler, checking the outflow of steam will usually stop it. If caused by dirty water, blowing down and pumping up will generally cure it. In cases of violent foaming check the draught and fires.

Air Leaks.—Be sure that all openings for admission of air to boiler or flues, except through the fire, are carefully stopped. This is frequently an unsuspected cause of serious waste.

Blowing-off.—Sludging should be frequently done, as it will tend to reduce incrustation. With muddy water, or salt water, the frequency of blowing-off should receive special attention, as the density of the water is then kept low.

Surface blow-off cocks are also advisable with muddy water, or salt water, or with water containing organic matter, and they should be used even more frequently than the blow-off cock.

Blow-off cocks should be examined every time the boiler is cleaned.

Leaks.—When leaks are discovered, they should be repaired as soon as possible, and as a new tube can be replaced in two or three hours, the use of clips or plugs should be avoided if a tube gives out. Never tighten nuts of handhole fittings whilst the boiler is under pressure.

Emptying Boiler.—Never empty the boiler while the brickwork is hot.

Filling up.—Avoid pumping cold water into a hot boiler.

Dampness.—Take care that no dampness can attack the exterior of the boiler.

If any pitting or redness is observed in times of boiler-cleaning, lime-washing the surfaces, and thoroughly cleaning them at regular intervals, will generally arrest the progres of corrosion.

Standing Idle.—If a boiler is not required for some time, empty, and dry thoroughly. If this is impracticable, fill it quite full of water, and put in a quantity of common washing soda.

External parts exposed to dampness should receive a coating of linseed-oil. The siphon pipe to steam-gauge, and connections to water-gauges, should always be drained.

Frost.—Do not leave water in a boiler when standing idle during frosty weather. When a boiler is standing idle in winter the water-gauges and steam-gauge should be examined to make sure that the passage-ways are clear.

General Cleanliness.—All things about the boiler-room should be kept clean and in good order.

Covering of Boilers.—The importance of covering the exposed parts of boiler shells and steam-pipes will be clearly seen from the following table. The loss of heat is further increased by the action of condensed water in the cylinders of an engine.

Below is given a table of the relative value of non-conducting materials. Magnesia covering protected by canvas is mostly used in electric-lighting stations, but some engineers prefer fibrous asbestos, owing to its being mechanically stronger.

Substance.	Value.
Felt, hair, or wool	1
Fossil meal	.66 to .79
Carbonate magnesia	.67 to .76
Brick	.70
Cork	.71
Loam, dry and open	.55
Asbestos paper	.47
Asbestos, fibrous	.36
Clay, with vegetable fibre ...	.34

CHAPTER VIII

MECHANICAL STOKERS

The utility of a stoker consists primarily in the mechanical feeding at regular intervals of such small quantities of coal that the time between each charge suffices for the gases to effect complete combustion; secondly, the avoidance of loss by the admission of excess of air through opening the fire-door when firing by hand; and thirdly, the reduction in human labour, which finds practical effect in one man being able to control a great number of furnaces or boilers, when mechanical stokers are added. Bituminous coal suits mechanical stokers, and their application is unquestionably the most correct method of ensuring good combustion with this fuel.

It is proposed to describe four kinds of stokers, viz.: Vicars, Bennis, Chain-Grate, and Meldrum's.

Vicars Stoker.—The Vicars Patent Stoker is of the coking type (fig. 1273).

The machine rests on the ground entirely, not hung on studs from the boiler front. Arrangements, however, are made so that the stoker can move freely on its supports as the boiler expands and contracts. The fuel is fed from the hoppers into boxes, usually two to each flue, from which it is pushed by reciprocating plungers or rams on to the dead-plate, where coking takes place. From the dead-plate it is pushed on to the moving fire-bars, which gradually take the burning mass farther into the flue. The partially consumed coke, with the clinker and ash, are discharged over the end of the fire-bars into the flue, where they form and maintain

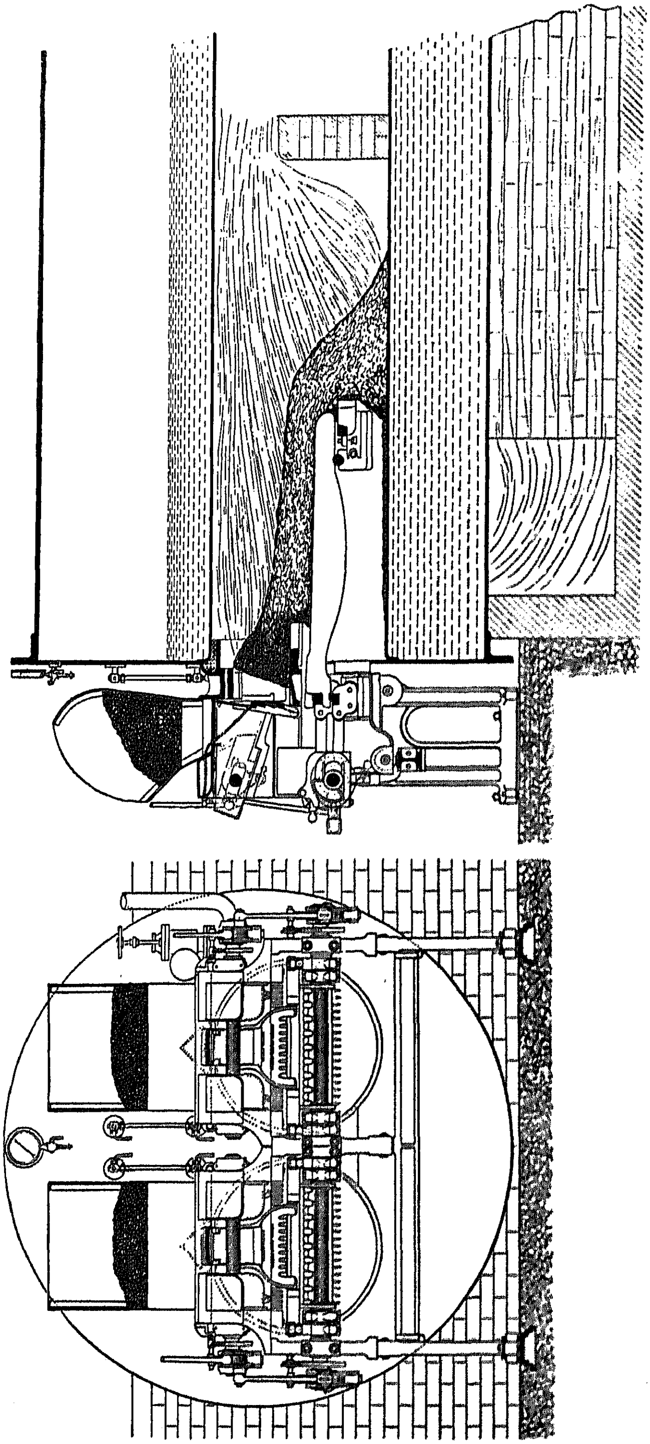


Fig. 1273.—Vicars Stoker

a bank, which acts as a bridge, and prevents any free air passing direct into the flue. Combustion is completed at this point, and the clinkers are removed at intervals according to conditions. A bridge of fire-brick is built in the furnace flue some feet from the end of the bars, so as to form a sort of combustion chamber. The supply of fuel and the travel of the bars are regulated with facility and independently of each other, and the gear for working each flue is entirely separate. The coal feed is varied by altering the rate of motion of the plungers, which, by a simple movement of a short lever, can be instantly adjusted to a wide range, or stopped altogether. The movement of the bars can also be varied. The bars of each furnace are arranged in two sets, each composed of the alternate bars, and move together when travelling in towards the bridge, but return at separate intervals. Thus the fuel is carried inwards by the simultaneous action of both sets of bars, and remains in place without being disturbed by the return of either set. Each successive inward movement of the bars serves to carry the fuel, together with the clinker and ash, nearer to the end of the grate, where the mass at length drops over the end of the bars, as already described.

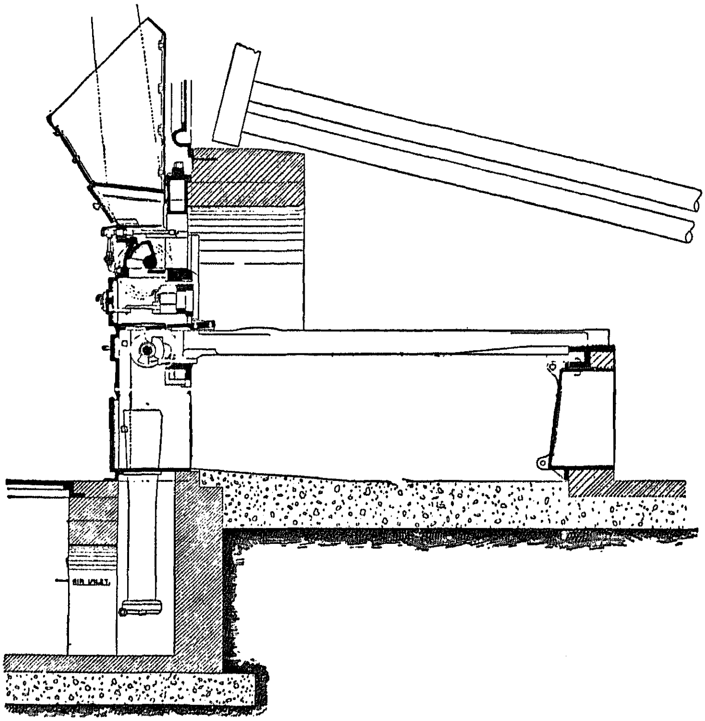


Fig. 1274.—Bennis Stoker

The Bennis Patent Stoker is of the sprinkling type (fig. 1274). In this machine the fuel is thrown, or fed by the conveyor or elevator, into hoppers, holding about three cwts., of which there are two to each Lancashire boiler. Under each hopper is placed a cast-iron feeding-box, in the interior of which there is a very simple pusher plate. A certain quantity of fuel falls in front of this plate, and is pushed by its movement over a ledge formed by the bottom of the box. The weight of the fuel so pushed over is regulated by means of a cam on the driving-shaft, so that the fireman knows how much fuel he is feeding, by the position of the cam. While the machine is running, the feed may be graduated from nothing up to a ton an hour. The fuel thus pushed over falls on a flat plate, which is called the shovel-box, from which it is projected into the fire by a shovel, propelled forward, striking the fuel and scattering it.

The shovel is actuated by a patent pneumatic gear. This consists of a long coiled spring, enclosed in a cylinder, and pressing on a piston. The spring is used to propel the shovel forward, and the air-cushion behind the piston prevents shock on the boiler front.

A cam draws back the shovel and has four varying lifts, the effects of this motion being to scatter the fuel on the fire in four divisions, each about eighteen inches long, so that in a six-feet furnace the fuel is thrown on only a quarter of the fire at once—a most material point where smokelessness is important, giving time for each portion of the fire to become incandescent between the charges.

When using low-class or waste fuels, which generally contain a large proportion of clinker and ash, the air-spaces in the fire-bars or ordinary furnaces become more or less covered and stopped up, and the fire suffers in consequence.

It is manifestly impossible to adjust the supply of air to the fire, to consume the fuel perfectly, unless the clinker and ash are continuously removed.

In the Bennis Furnace it is claimed that this is effectually accomplished. This consists of tubular fire-bars of the length the fire is intended to be, which all move into the fire together about two inches, and are then drawn out one by one by the action of cams placed on a shaft. These cams are made the full width of the bar (about four inches or so), so that there is little wear upon them.

The clinker and ash are slowly carried forward by this action to the end of the bars, where they drop over into a closed chamber, give up their heat to the boiler, and are drawn out at intervals.

The air-spaces between the bars being always free and open, and each tubular bar having its own supply of compressed air, it is claimed that the draught is evenly distributed over the whole fire-grate, and the boiler continues to do its work, while cleaning out the clinker from the chamber.

The bars are constructed with small air-spaces, so that breeze or dust fuel may be burned with advantage. Any wear of the cams upon the compressed air-bars is taken up by adjustable, chilled cast-iron wearing pieces. The wearing parts of the moving furnace-bars can be adjusted or replaced, and the small interlocking bars subject to the action of the fire may also be changed when worn out; the fire should never reach the tubular fire-bars.

Chain Grate Stoker.—This machine is of the coking type, and is illustrated by figs. 1275 and 1276. The coal is fed into a hopper as in the other machines described, and falls from thence on to a travelling grate.

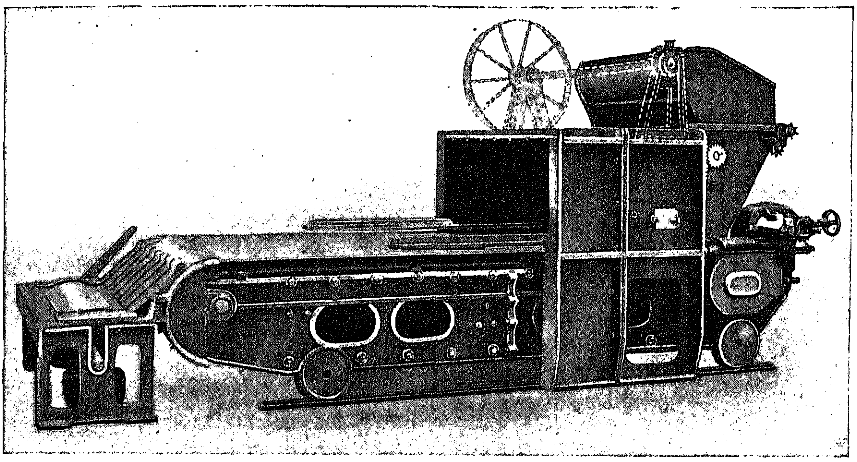


Fig. 1275.—Chain-Grate Stoker—rear view

This grate is practically an endless chain, the links of which form the grate bars. A brick arch is built over the front portion of the grate, and

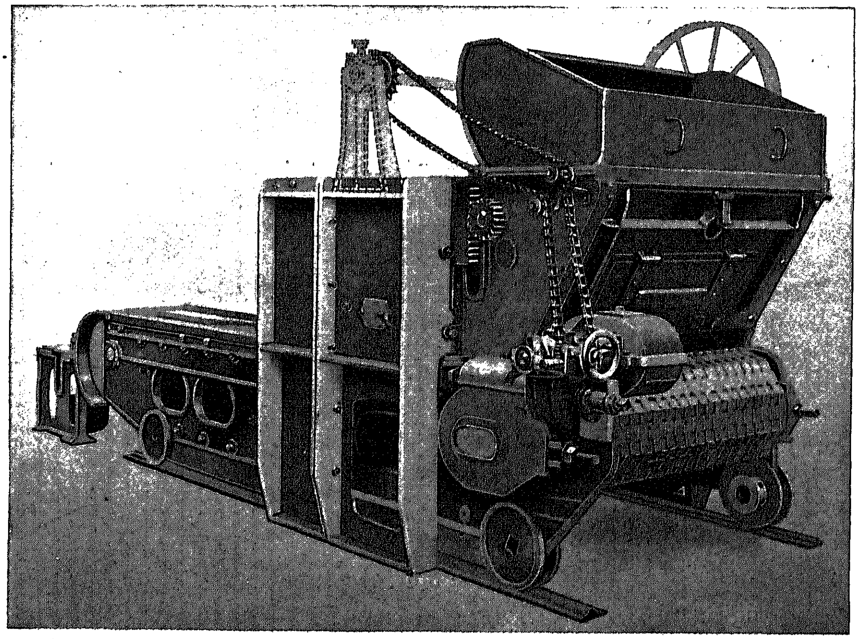


Fig. 1276.—Chain-Grate Stoker—front view

becomes highly heated in working; as the fuel is carried into the furnace it becomes coked, and the gases leaving it are ignited by the hot arch above it. The links are of a special interlocking pattern, which allow the smallest coals to be used, and, while providing for an ample supply of air for the combustion of the fuel, prevent the influx of cold air at the back of the grate. The starting or stopping of the stoker, or the variation of speed, is effected through the gear box. This box contains an efficient clutch for putting the driving gear out of action should any unusual strain arise.

The speed at which the grate travels and the thickness of the fire can be adjusted by simple means, according to the demand for steam. The speed of the grate must be such that the fuel is consumed by the time it reaches the back of the furnace.

At this point the ash and clinker fall over into a chamber provided for them. The latter can be emptied from time to time by a lever outside the boiler walls.

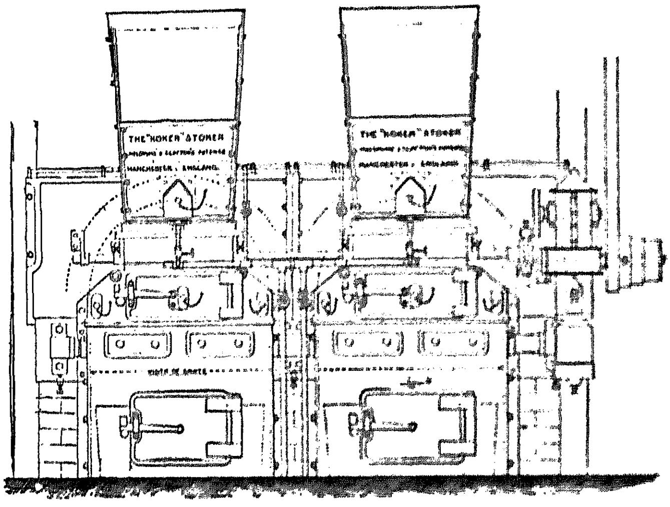


Fig. 1277.—Meldrum's "Koker" Stoker

Provision is made in this stoker for hand-firing in the event of any break-down in the driving mechanism.

This stoker is one of the simplest and strongest made; the details have been so much improved of recent years that there is little in common between it and its prototypes.

Meldrum Stoker.—Fig. 1277 illustrates the front elevation and longitudinal section of the "Koker" stoker, as applied to a water-tube boiler; the forced-draught features of the machine are clearly shown. The vertical position of the blowers has been adopted, in order that the air supply may be delivered direct to the cams and bar-ends. This increases their durability as the cooling action facilitates lubrication. It will be noticed in the longitudinal section that there are two fire-doors, an outer one which is solid and an inner one which is barred and shackled to the outer one, so that they open simultaneously. The intermediate space is in communication with auxiliary steam-jet blowers in the machine front, and receives the forced supply of air independent of the ash-pit supply. When the stoker is in operation the fuel falls from the hoppers in front of the sector-shaped rams, which move through variable angles according to the duty required. This movement sends the fuel forward over the sill until it banks in front of the inner fire-door, and comes under the influence of the moving grate. The grate movement is effected by a series of cams on a hexagonal shaft which engage with the bar-ends. Two types of bar-ends are used, and these are arranged alternately. The revolutions of the hexagonal shaft cause both types of bars to move towards the clinker pit simultaneously, but they return alternately. The fuel is thus conveyed to the clinker pit.

The coking action which gives this stoker its name occurs when the fuel is banked at the front of the machine, and subject to the radiant heat of the more active portion of the grate. The gases evolved are consumed by the air delivered by the auxiliary blowers previously referred to, and it is claimed that this is an essential feature of the machine as a smoke preventer.

CHAPTER IX

SUPERHEATING

Definition.—Steam is said to be superheated when at any given boiler pressure it has a higher temperature than the water from which it was evaporated. In electric light and power stations in this country the practice of superheating steam is becoming almost universal, and although there is little exact data published of British practice, there is no doubt that very considerable saving has been made.

The object in superheating is to prevent the formation of water by condensation in steam-pipes and in the cylinders of the engines. It has been stated that in an ordinary single-cylinder, non-jacketed, non-condensing engine, the amount of waste from condensation is 30 to 40 per cent of the steam used; but Mr. Lenke, in a paper read before the Institution of Mechanical Engineers, 1901, stated the condensation of steam during admission as 20 to 25 per cent. Bryan Donkin, however, thought this too high, except in single-cylinder, non-jacketed engines.

By superheating steam these losses may be avoided to a great extent. This fact has long been recognized, and on the Continent superheating has been practised to a far greater extent than in this country. The chief causes of the slow progress of superheating are that there was, until recent years, no reliable apparatus on the market. Further, old-fashioned animal and vegetable lubricating oils were destroyed by the high temperature before they had a chance to do their work, and the stuffing-box packings were charred. The advent of high-pressure steam, and the consequent increase of temperature, have caused mineral oils and metallic steam packings to be almost universally adopted. The old difficulties in these respects of the pioneers of the fifties have consequently disappeared, and it only remained for the inventor to design a reliable apparatus.

The superheaters which will be described are those of the Babcock and M'Phail types, but before doing so it will be well to state the principles upon which a modern superheater should be constructed.

It should be placed in a position where a suitable temperature is available. It should be constructed to admit of all its parts expanding and contracting freely, without severe strains being put upon any of the joints, which might cause them to leak; flanged joints, when exposed to furnace gases, should be avoided wherever possible. Provision should be made to protect the superheater against overheating when sufficient steam is not passing through it to carry away the heat supplied by the furnace gases.

Regarding the first point there is much diversity of opinion, and a good deal depends upon the type of boiler. Suppose, for instance, a boiler is working economically at a gauge pressure of 100 lbs. per square inch, the temperature of the saturated steam would be 338° F. If the temperature of the gases leaving the boiler was only 400° F. the difference of temperature between the steam and the gases would be only, say, 62 degrees; and it is obvious in this case that it would be useless to place the superheater in any position between the boiler dampers and the chimney. It may be taken as a general rule that a superheater should not be placed in such a position. On the other hand, the superheater must not be placed too near the furnace, as otherwise excessive heat may damage the apparatus. If the superheater is not of the separately-fired type it is usually, in the case of a Lancashire boiler, set in the back flue. In the case of the Babcock & Wilcox boiler, where the superheater forms an integral part of the boiler, it is set between the top of the inclined tubes and the steam-and-water drum. In each of these cases the superheater is not subject to the immediate action of the fire, as the furnace gases, while still retaining a considerable quantity of heat, have parted with not a little to the boiler-heating surface.

The M'Phail & Simpson superheater consists primarily of two superheaters set in the back flue. The saturated steam generated in the boiler passes through the first superheater, is then carried back by copper pipes through the water of the boiler, and is then returned to the second superheater, where it is again superheated, and from thence it passes to the engine. In passing back through the boiler it does duty in evaporating water, so that it performs a double function—that of superheater and evaporator. Whether this is an advantage or not is a question of experience, but it must undoubtedly increase the efficiency of the boiler. Apparently, however, the same result could be arrived at by increasing the heating surface of the boiler and carrying the steam, when superheated, directly away to the steam cylinder. It is, however, sometimes installed in places where there is neither room nor time to put in additional boiler power.

The Babcock & Wilcox superheater applied to a Lancashire boiler, and also independently-fired superheaters of the same make, are illustrated and described below. In the type applied to their own boilers (fig. 1278) the tubes are bent into a **U** shape, and connected at both ends with manifolds, one of which receives the saturated steam from the boiler, the other collecting the steam after it has traversed the superheated tubes, and delivering it to the valve placed above the boiler.

The flooding arrangement (fig. 1279) consists in a connection with the water-space of the boiler drum, by means of which the water enters the lower manifold and fills the superheater to the boiler water-level. Prior to opening the superheated stop-valve and using superheated steam, the water is drained away from the manifolds by the flooding pipe and cock. A sight-glass is attached to the drain outlet for ascertaining that all the water has been drawn out of the superheater when draining.

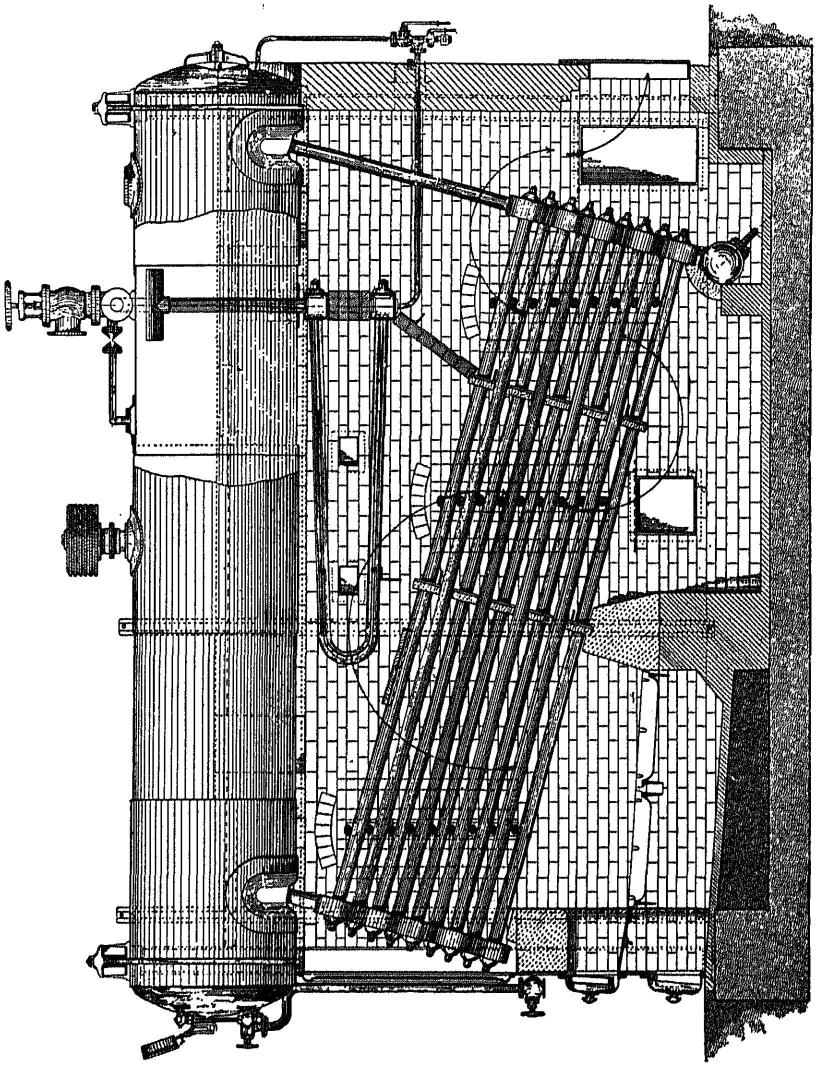


Fig. 1278.—Babcock & Wilcox Boiler, with Superheater

Superheaters are sometimes fitted to Lancashire boilers; they are placed at the back of the boiler where the gases leave the internal flues on their way to the external ones.

They consist of a number of steel tubes of small diameter, bent to a U form. The ends of the tubes are expanded into steel manifolds, one of which is connected to the steam space of the boiler and the other to the main steam-pipe.

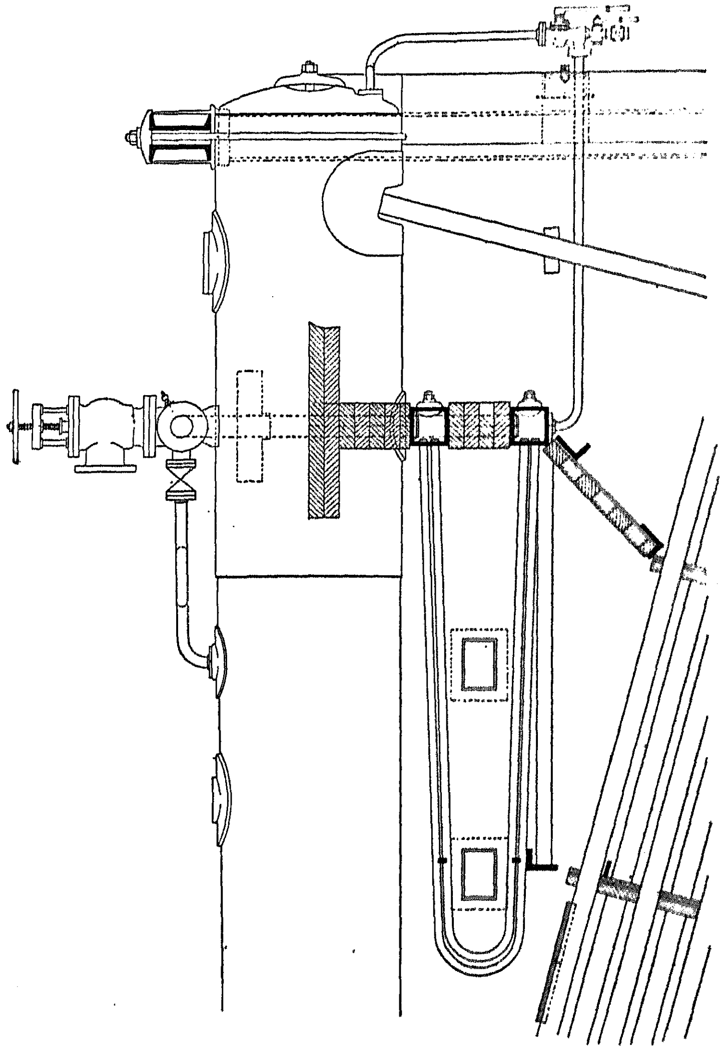


Fig. 1279.—Flooding Arrangement of Babcock & Wilcox Superheater

It will be seen from the illustration (fig. 1280) that the superheater can be cut out of work by dampers, which permit of the gases being diverted directly to the external flues of the boiler without passing over the superheater tubes. An arrangement of valves is also provided, by which the steam can be passed direct from the boiler to the steam main without passing through the superheater.

The separately-fired type of superheater, illustrated by fig. 1281, is composed of a series of boxes or manifolds connected together by **U**-shaped tubes. The steam from the steam main enters the first box and is conveyed through the tubes into the second, third, and fourth boxes, and so on, passing from the last box back into the steam main.

The gases generated on the grate rise upward, passing through a perforated wall into the superheater chamber, and thence through a similar perforated wall into the chimney. The amount of heat passing through the superheater chamber can be regulated at will. The temperature is kept comparatively low—the grate surface required being small.

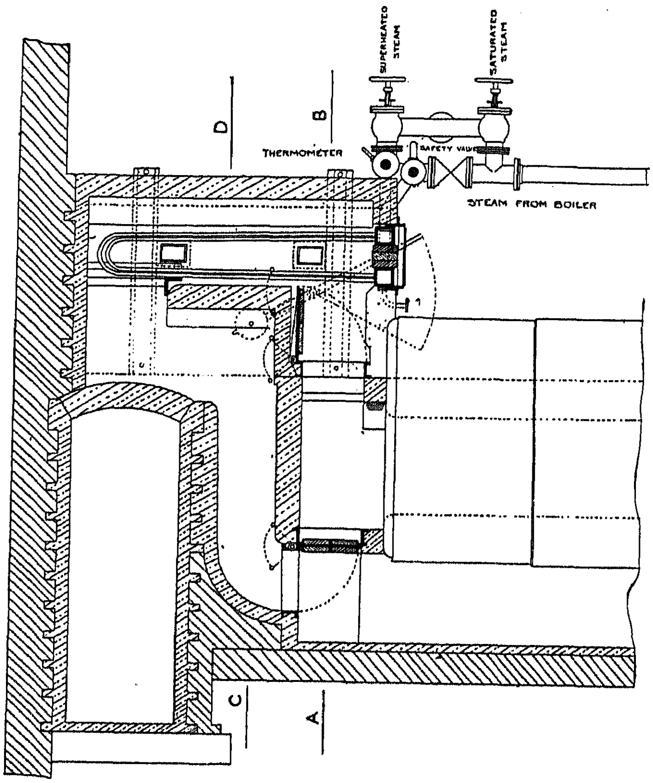


Fig. 1280.—Superheater fitted to Lancashire Boiler

The superheater is provided with a by-pass in case of any necessity for repairs, and also with the necessary valves to enable it to be shut off from the steam main if desired without interfering with the general running of the plant. The quantity of fuel required is exceedingly small.

General.—In English practice from 100° to 150° F. of superheat is usually provided for; and it has been found by experiment that even with the very best triple-expansion high-speed engines, working under ordinary conditions, an economy of from 10 to 15 per cent can be regularly obtained. With engines less refined, or of slower piston speed, the percentage of saving is greater.

On the Continent, steam superheated 250° to 260° F. is often employed, and numbers of superheaters are working under such conditions most satisfactorily.

Professor Schroter has carried out some very careful tests on a triple-expansion engine, using with saturated steam only 12 lbs. per hour, and therefore of the highest class. With the steam superheated to 500° F. the consumption was reduced by 18 per cent. Even higher economies have been effected with engines of a less efficient type, Messrs. Hick, Hargreaves, & Co. having found a gain of 30 per cent with simple engines working at 60 lbs. pressure, and 23 per cent with triple-expansion engines working at 200 lbs. pressure. Mr. Michael Longridge is of opinion that the saving in the case of a good mill engine, with a superheat of 100° to 120° F., is about 10 to 12 per cent.

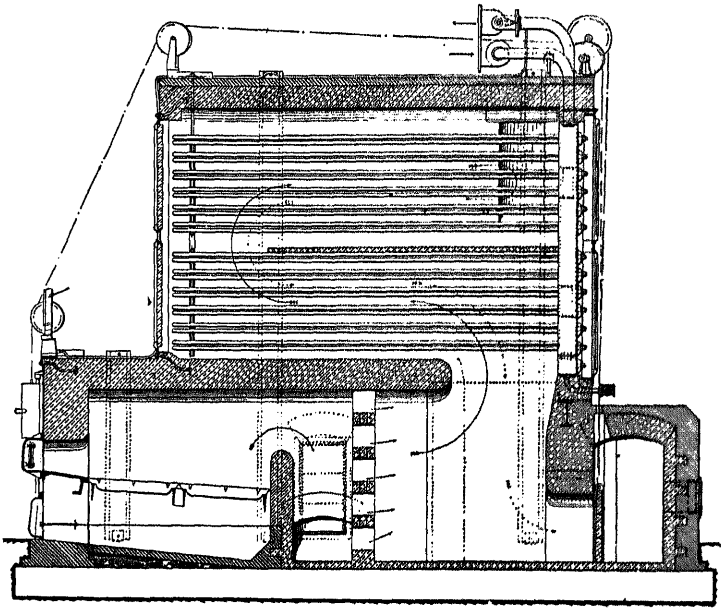


Fig. 1281.—Separately-Fired Superheater

The saving obtained by superheating has been borne out in a practical manner by the records of several electric-light stations; and in one notable instance, at Bournemouth, from records taken over two years on the whole boiler plant, a saving of 15 per cent was made on the first year and 20 per cent the second year, allowance being made for any slight difference in conditions—the engine and boiler plant being the same, and the class of fuel the same also.

The principal reasons for the economy shown by the use of superheated steam may be briefly summarized as follows:—

(1) Initial condensation in the engine cylinders is reduced. The experiments of the late Bryan Donkin proved what had been theoretically assumed previously—that when dry steam (which is, of course, invisible)

is admitted to the cylinder, the cooling action of the walls produces a mist, which settles on them in minute drops of water; immediately the valve closes and expansion begins these drops are re-evaporated. These actions represent a direct loss of heat at every stroke of the engine, which can be avoided if the steam is sufficiently hot to begin with, so that the walls do not cool it down to condensation point.

(2) A reduction is effected in the weight of steam required to fill the steam passages and clearance spaces, owing to the greater volume of the steam when superheated. Closely allied to this is the reduced leakage past valves and pistons.

(3) A great saving is effected by the avoidance of condensation in the steam-pipes leading from the boilers to the engines. Superheated steam does not part with its heat so readily as saturated steam. An analogy may be found in connection with our own bodies: we do not feel the cold nearly so much on a clear sharp frosty day as we do when the temperature is considerably higher, but the air is charged with moisture.

A word of caution may be given as to steam-pipes when superheated steam is used. Greater provision must be made for expansion, owing to the higher temperature, than with saturated steam, and copper pipes must not be employed. Copper having a tenacity of 31 tons at 60° F. is reduced to 11 tons at 482° F., and it is consequently quite unsuitable for bends. Care must also be taken that the seats of stop-valves are designed in such a way that the variations in temperature do not cause them to leak.

Below is given particulars of a test on a Babcock & Wilcox superheater fitted to a boiler of the semi-tubular type at Messrs. Cockerill & Co.'s works, Seraing, Belgium.

TESTS ON CONDENSING ENGINE NO. 2389, BELONGING TO MESSRS. COCKERILL & CO., AT SERAING,

Of the 22nd June, 1899, Boiler without and with Superheater

Dimensions of the Engine

High-pressure cylinder diameter	15.4	inches.
Low-pressure cylinder diameter	22	„
Stroke	30.4	„
Diameter of piston-rod	2.4	„

Dimensions of the Boiler (Semi-Tubular)

Heating surface	570.75	sq. ft.
Surface of fire-tubes of 90 millimeters diameter	118	„
Diameter of shell	4.592	ft.
Length of shell	11.8	„
3 "Bouilleurs" (small drums) diameter	24.8	in.
3 "Bouilleurs" (small drums) length	92	„
Grate width	55.6	„
Grate length	52.32	„
Grate surface	19.35	„

Babcock & Wilcox Superheater

Number of tubes 9 × 4 inches … … … … 36
Heating surface … … … … … … 92.2 sq. ft.

	Test on 22nd June without superheater.	Test on 22nd June, 1899, with superheater.
Duration of test	5 h.	5 h.
Revolutions per minute	51.96	54.12
Mean I.H.P.	41.87	48.20
Total feed-water	6094.22 lbs.	6320.5 lbs.
Total feed-water per hour	1218.8 „	1264.09 „
Mean pressure	69.5 „	60.27 „
Coal fired gross	792 „	754 „
Ash	34.1 „	61.6 „
Net coal	757.9 „	692.4 „
Net coal per hour	151.6 „	138.5 „
Evaporation per lb. of coal	8 „	9.12 „
Mean temperature flue gases at the base of chimney	475° F.	500° F.
Mean temperature of the steam before superheating		302° F.
Mean temperature of the steam after superheating		384° F.

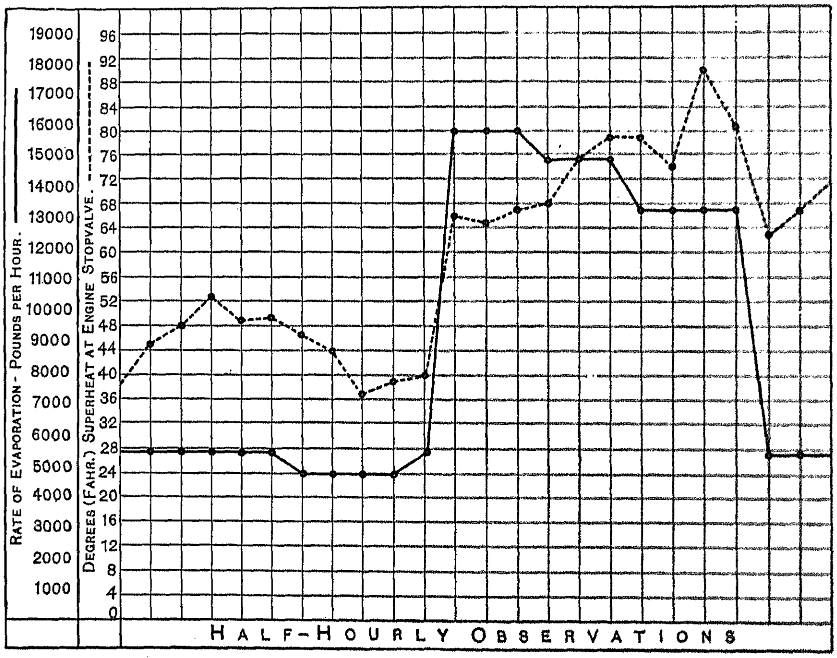


Fig. 1282

A diagram of curves is given in fig. 1282, showing the results of a series of half-hourly observations to ascertain the relation between the amount of superheat and the rate of evaporation obtained from the Babcock & Wilcox boiler with superheater, indicating the self-regulation of superheater, at the Reading electric-light station.

CHAPTER X

CHIMNEYS

Draught.—Chimneys are required for two purposes: first, to carry off obnoxious gases; second, to produce a draught, and so facilitate combustion.

Each pound of coal burned yields from 13 to 30 pounds of gas, the volume of which varies with the temperature.

The weight of gas carried off by a chimney in a given time depends upon three things—size of chimney, velocity of flow, and density of gas. But as the density decreases directly as the absolute temperature, while the velocity increases with a given height nearly as the square root of the temperature, it follows that there is a temperature at which the weight of gas delivered is a maximum. This is about 550 degrees above the surrounding air. Temperature, however, makes so little difference, that at 550 degrees above, the quantity is only 4 per cent greater than at 300 degrees. Therefore height and area are the only elements necessary to consider in an ordinary chimney.

The intensity of draught is, however, independent of the size, and depends upon the difference in weight of the outside and inside columns of air, which varies nearly as the product of the height into the difference of temperature. This is usually stated in an equivalent column of water, and may vary from 0 to possibly 2 inches.

After a height has been reached to produce draught of sufficient intensity, provided the area of the chimney is large enough, there seems no good mechanical reason for adding further to the height, whatever the size of the chimney required. Where cost is no consideration there is no objection to building as high as one pleases; but for the purely utilitarian purpose of steam-making, equally good results might be attained with a shorter chimney at much less cost.

The intensity of draught required varies with the kind and condition of the fuel, and the thickness of the fires; $\frac{3}{8}$th inch gives good average results.

A round chimney is better than a square, and a straight flue better than a tapering, though it may be either larger or smaller at top without detriment.

The effective area of a chimney for a given power varies inversely as the square root of the height. The actual area in practice should be greater, because of retardation of velocity due to friction against the walls.

The following formula gives the draught of a chimney in inches of water, where d = draught, h = height of chimney in feet, τ = absolute temperature of air, t = absolute temperature of gases:—

$$d = h\left(\frac{7.6}{\tau} - \frac{7.9}{t}\right)$$

The height of a chimney, to give a specific draught power expressed in inches of water, may be found as follows:—

$$h = \frac{d}{\left(\frac{7.6}{T} - \frac{7.9}{t}\right)}$$

The external diameter of a brick chimney at the base should be one-tenth the height, unless it is supported by some other structure. The "batter" or taper of a chimney should be from $\frac{1}{16}$ to $\frac{1}{4}$ inch to the foot on each side.

Thickness of brickwork—one brick (9 inches) for 25 feet from the top, increasing half brick ($4\frac{1}{2}$ inches) for each 25 feet from the top downwards.

If the inside diameter exceed 5 feet the top length should be $1\frac{1}{2}$ brick, and if under 3 feet it may be $\frac{1}{2}$ brick for 10 feet.

SIZES OF CHIMNEYS FOR BOILER PLANTS OF A GIVEN EVAPORATION

Diameter in inches.	Height of Chimneys for Evaporative Capacity (in lbs. per hour) of Boiler Plants.											Side of Square.	Effective Area.	Actual Area.
	50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.	ins.	sq. ft.	sq. ft.
18	700	800	900									16	0.97	1.77
21	1100	1200	1300									19	1.47	2.41
24	1500	1700	1800	1,900								22	2.08	3.14
27	2000	2200	2400	2,500								24	2.78	3.98
30	2600	2800	3000	3,300	3,400							27	3.58	4.91
33		3500	3800	4,000	4,300							30	4.48	5.94
36		4300	4600	4,900	5,200	5,500						32	5.47	7.07
39			5500	5,900	6,300	6,600						35	6.57	8.30
42			6500	7,000	7,400	7,800	8,200					38	7.76	9.62
48				9,400	9,900	10,500	11,000	11,700				43	10.44	12.57
54				10,900	12,900	13,500	14,200	15,100	16,600			48	13.51	15.90
60				15,200	16,100	17,000	17,800	19,000	20,800	22,500		54	16.98	19.64
66					19,800	20,900	21,900	23,300	25,500	27,600	29,500	59	20.83	23.76
72					23,800	25,100	26,300	28,100	30,700	33,200	35,500	64	25.08	28.27
78						29,900	31,200	33,300	36,400	39,300	42,000	70	29.73	33.18
84						34,900	36,500	38,900	42,600	46,000	49,200	75	34.76	38.48
90						40,400	42,500	44,900	49,200	53,100	56,800	80	40.19	44.18
96						46,200	48,500	51,600	56,300	60,900	65,100	86	46.01	50.27
102								58,400	64,000	69,100	73,900	90	52.23	56.75
108								65,800	72,100	77,900	83,200	96	58.83	63.62
114								73,800	80,700	87,100	90,100	101	65.83	70.88
120									89,700	96,900	103,600	106	73.22	78.54
126									99,300	107,200	114,600	112	81.00	86.59
132									109,300	118,100	126,200	117	89.19	95.03
138									119,800	129,400	138,300	122	97.75	103.86
144									130,800	141,300	151,000	127	106.72	113.10

CHAPTER XI

COMBUSTION AND EVAPORATION

No work on boilers would be complete without some reference to the theory of combustion.

With one exception, all the elements enter into chemical combination with oxygen, forming new compounds called oxides. When the combination is attended with light and heat they are said to *burn* or undergo combustion, and this is the process which takes place in the furnace of a steam-boiler. The various elements forming the fuel enter into combination with the oxygen of the air, and the heat derived from the process is utilized to evaporate the water in the boiler into steam.

The amount of heat produced by the combustion of different bodies has been established by experiment, and is expressed in heat units. A heat unit (H.U.), or, as it is sometimes called, British thermal unit (B.T.U.), is the amount of heat required to increase the temperature of 1 lb. of water by 1° F. If exactitude is required, the water must be taken at its greatest density, *i.e.* at a temperature of 39.1° F.

The value of a fuel as a steam producer can therefore be directly stated in H.U. per pound of the fuel. This value may be arrived at by analysing the fuel and calculating the total heating value of its constituents, or it may be ascertained directly by experiment.

The principal constituents of fuel are carbon, oxygen, and hydrogen, in varying proportions, with a small amount of sulphur. The latter may generally be neglected in calculations of the heating value of the fuel for all practical purposes, as its amount is insignificant, although it is sometimes met with in appreciable quantities in the form of pyrites. There are also other bodies in fuel, such as nitrogen, water, and certain mineral matters which constitute the ash. These take no part in the production of heat, and consequently have not to be considered in calculating the calorific value of a fuel, although for other purposes of the analysis they may be of interest.

The carbon is the principal heat-producing constituent, and it has been established by many careful and delicate experiments that 1 lb. of this element, when completely burned, produces 14,500 H.U.

Hydrogen is far richer in heat than carbon, producing 62,000 H.U. per pound; but there is a very small proportion of this element in any fuel compared with the carbon, and a large amount of what there is exists in combination with the oxygen in the form of water. Every eight parts by weight of this oxygen require one part of hydrogen, and until all the oxygen is satisfied, none of the hydrogen is available for the purpose of steam raising. The water formed by the combination of these two elements is driven off as steam in the furnace, and it condenses again into water in the flues or after it has escaped from the chimney.

In calculating the heat available from the hydrogen, therefore, one part for every eight parts of oxygen must be deducted.

The heating power of a compound body being equal to the sum of the heating powers of its constituents, we can determine the value of a fuel as a heat-producer when its constituents are known.

Let C, H, and O represent the percentages of carbon, hydrogen, and oxygen respectively;

(1) Then $\frac{14{,}500 \times C}{100} =$ heating value of the carbon.

$$(2)\ \frac{62{,}000\left(H - \frac{O}{8}\right)}{100} = \text{heating value of the hydrogen.}$$

Let *h* be the heating value of the fuel; then,

$$(3)\ h = \frac{14{,}500\ C + \left(14{,}500 \times 4.28\left\{H - \frac{O}{8}\right\}\right)}{100}.$$

$$(4)\ \text{or } h = 145\left(C + 4.28\left\{H - \frac{O}{8}\right\}\right).$$

In order to make clear the process, we will take an actual analysis as an example. The coal analysed was ordinary Lancashire engine slack, and was composed of the following elements, in the proportion stated:—

Carbon	74.62	per cent
Hydrogen	4.20	„
Nitrogen	1.52	„
Oxygen	10.65	„
Sulphur	.36	„
Ash	8.65	„
	100	

By formula 4,

$$\begin{aligned} h &= 145\left(74.62 + 4.28\left\{4.2 - \frac{10.65}{8}\right\}\right) \\ &= 12{,}600. \end{aligned}$$

The total heating power of the fuel cannot, however, be all utilized in any boiler, as there are certain losses, some of which are unavoidable, and others which, although they may be minimized by care in the design and working of the boiler, yet cannot be avoided altogether. Among the latter, one of the most important is that arising from incomplete combustion.

Carbon combines with oxygen to form two distinct gases, carbonic acid (CO_2) and carbonic oxide (CO).

The former is composed of one part by weight of carbon and two parts of oxygen, while the latter is composed of equal parts of these elements. 1 lb. of carbon when forming CO_2 supplies 14,500 H.U.; when forming CO, only 4450 H.U. Further, the volume of CO is double that of an equal weight of CO_2, and a large amount of heat is absorbed in expanding the gas. It is evident, therefore, that it is exceedingly wasteful to use our carbon in forming CO rather than CO_2. If an ample supply of air is provided, the CO will take up a further portion of oxygen, forming CO_2; the combustion is completed, and the full heating value of the carbon is utilized. It is therefore of the utmost importance that the furnace should be so designed that an ample supply of air is admitted. Care must, of course, be taken that this is not carried to excess, for in that case the air reduces the temperature of the gases below that required for complete combustion and the efficiency is impaired, while a large amount of black smoke will be produced.

The theoretical amount of air required for combustion may be determined from an analysis of the fuel by the formula

$$(5)\quad 12\,C + 36\left(H - \frac{O}{8}\right) = \text{lbs. of air required per lb. of fuel,}$$

C, H, and O being the weight percentages of the elements as in formula 4.

In practice, however, at least double this quantity is required with natural draught. If forced draught is used the quantities may be reduced by 20 to 25 per cent.

The following table gives the average heating value of British coals. Coals, however, vary greatly in composition in any given district, and if an accurate test of a boiler is to be made, the coal used should be specially analysed.

TABLE II.—CALORIFIC VALUES OF DIFFERENT SAMPLES OF COALS

	B.T.U.	Calories.	
Welsh.			
Lewis Merthyr	15,550	8639	
Powell Duffryn	15,702	8723	
Ebbw Vale	15,620	8678	
Penrikyber	15,500	8611	
Fforchamen	15,200	8445	
English.			
West Hartley	12,358	6866	
Acton Hall	12,541	6967	
Newcastle	12,934	7185	
Manners	13,306	7392	
Wheldale	13,509	7505	
Scotch.			
Whistlebury	13,402	7445	
Dechmont	12,902	7168	73 per cent carbon
Bothwell	12,694	7052	
Over Dalself	12,367	6871	
Aitken	13,770	7650	85.5 per cent carbon.

Assuming that a supply of heat has been obtained, the next step is to convert it into mechanical energy. The simplest and most universally applicable method of doing this is by evaporating water and using the resulting steam to drive an engine.

If we take 1 lb. of water in an open vessel and supply heat to it, we raise its temperature 1° F. for every heat unit added until we reach 212°. After that temperature is reached, we may go on adding heat but the water does not get any hotter; steam begins to form, and when we have added 967 heat units, the whole of the water will have been converted into steam. If we could prevent the steam escaping we should have 1 lb. of steam at a temperature of 212°.

If the water were contained in a closed vessel, and we continued to add heat, the steam formed would rise in both temperature and pressure. The following table gives the temperature and heat units contained in 1 lb. of steam at various pressures:—

TABLE OF PROPERTIES OF SATURATED STEAM

Pressure in lbs. per sq. in. above Vacuum.	Temperature in Degrees Fahr.	Total Heat in Heat Units from Water at 32°.	Heat in Liquid from 32° in Units.	Heat of Vaporization, or Latent Heat in Heat Units.	Density or Weight of cubic feet in lbs.	Volume of 1 lb. in cubic feet.	Factor of Equivalent Evaporation at 212°.	Pressure in lbs. per sq. in. above Vacuum.
1	101.99	1113.1	70.0	1043.0	0.00299	334.5	.9661	1
2	126.27	1120.5	94.4	1026.1	0.00576	173.6	.9738	2
3	141.62	1125.1	109.8	1015.3	0.00844	118.5	.9780	3
4	153.09	1128.6	121.4	1007.2	0.01107	90.33	.9822	4
5	162.34	1131.5	130.7	1000.8	0.01366	73.21	.9852	5
6	170.14	1133.8	138.6	995.2	0.01622	61.65	.9876	6
7	176.90	1135.9	145.4	990.5	0.01874	53.39	.9897	7
8	182.92	1137.7	151.5	986.2	0.02125	47.06	.9916	8
9	188.33	1139.4	156.9	982.5	0.02374	42.12	.9934	9
10	193.25	1140.9	161.9	979.0	0.02621	38.15	.9949	10
15	213.03	1146.9	181.8	965.1	0.03826	26.14	1.0003	15
20	227.95	1151.5	196.9	954.6	0.05023	19.91	1.0051	20
25	240.04	1155.1	209.1	946.0	0.06199	16.13	1.0099	25
30	250.27	1158.3	219.4	938.9	0.07360	13.59	1.0129	30
35	259.19	1161.0	228.4	932.6	0.08508	11.75	1.0157	35
40	267.13	1163.4	236.4	927.0	0.09644	10.37	1.0182	40
45	274.29	1165.6	243.6	922.0	0.1077	9.285	1.0205	45
50	280.85	1167.6	250.2	917.4	0.1188	8.418	1.0225	50
55	286.89	1169.4	256.3	913.1	0.1299	7.698	1.0245	55
60	292.51	1171.2	261.9	909.3	0.1409	7.097	1.0263	60
65	297.77	1172.7	267.2	905.5	0.1519	6.583	1.0280	65
70	302.71	1174.3	272.2	902.1	0.1628	6.143	1.0295	70
75	307.38	1175.7	276.9	898.8	0.1736	5.760	1.0309	75
80	311.80	1177.0	281.4	895.6	0.1843	5.426	1.0323	80
85	316.02	1178.3	285.8	892.5	0.1951	5.126	1.0337	85
90	320.04	1179.6	290.0	889.6	0.2058	4.859	1.0350	90
95	323.89	1180.7	294.0	886.7	0.2165	4.619	1.0362	95
100	327.58	1181.9	297.9	884.0	0.2271	4.403	1.0374	100
105	331.13	1182.9	301.6	881.3	0.2378	4.205	1.0385	105
110	334.56	1184.0	305.2	878.8	0.2484	4.026	1.0396	110
115	337.86	1185.0	308.7	876.3	0.2589	3.862	1.0406	115
120	341.05	1186.0	312.0	874.0	0.2695	3.711	1.0416	120
125	344.13	1186.9	315.2	871.7	0.2800	3.571	1.0426	125
130	347.12	1187.8	318.4	869.4	0.2904	3.444	1.0435	130
140	352.85	1189.5	324.4	865.1	0.3113	3.212	1.0453	140
150	358.26	1191.2	330.0	861.2	0.3321	3.011	1.0470	150
160	363.40	1192.8	335.4	857.4	0.3530	2.833	1.0486	160
170	368.29	1194.3	340.5	853.8	0.3737	2.676	1.0502	170
180	372.97	1195.7	345.4	850.3	0.3945	2.535	1.0517	180
190	377.44	1197.1	350.1	847.0	0.4153	2.408	1.0531	190
200	381.73	1198.4	354.6	843.8	0.4359	2.294	1.0545	200
225	391.79	1201.4	365.1	836.3	0.4876	2.051	1.0576	225
250	400.99	1204.2	374.7	829.5	0.5393	1.854	1.0605	250
275	409.50	1206.8	383.6	823.2	0.5913	1.691	1.0632	275
300	417.42	1209.3	391.9	817.4	0.644	1.553	1.0657	300
325	424.82	1211.5	399.6	811.9	0.696	1.437	1.0680	325
350	431.90	1213.7	406.9	806.8	0.748	1.337	1.0703	350
375	438.40	1215.7	414.2	801.5	0.800	1.250	1.0724	375
400	445.15	1217.7	421.4	796.3	0 853	1.172	1.0745	400
500	466.57	1224.2	444.3	779.9	1.065	.939	1.0812	500

Supposing that all the heat contained in the fuel could be utilized in converting the water into steam, the theoretical evaporation per pound of fuel from water at 212°, and at atmospheric pressure (or, as it is called, from and at 212°), is as follows:—

$$E = \frac{C}{966.6}$$

where E is the theoretical evaporation and C is the calorific value of the fuel.

Certain losses, however, are unavoidable in the best boilers, and the actual evaporation in practice is about 70 per cent of the above, so that the evaporation per pound of coal of any known quality which may be expected from a boiler of good design is as follows:—

$$e = \frac{C \times 70}{966.6 \times 100}$$

where e is the actual evaporation.

The efficiency of any boiler, when the evaporation per pound of fuel is known from a test, is obtained by transposing, thus:—

$$\text{Efficiency} = \frac{966.6 \times e \times 100}{C}.$$

The above formulæ assume evaporation from feed-water at 212° and at atmospheric pressure, and it is convenient to reduce the results of all tests to this basis, so as to eliminate differences introduced by varying steam pressure and temperature of feed-water.

The relation between this "standard evaporation", as it is called, and that actually observed, is expressed by the following formula:—

$$E = e\frac{(h + 32 - t)}{966.6}$$

where e = actual or observed evaporation, h is the total heat contained in the steam, as given in table, and t = temperature of the feed-water.

CHAPTER XII

TESTING BOILERS

In order to ascertain the efficiencies of steam-boilers, or their maximum evaporative capacity under specified conditions, it is necessary to carry out somewhat elaborate experiments, in which great care is necessary if reliable results are to be obtained.

If it is desired to know the most economical rate of working the boiler, then one experiment will not suffice, and it will be found that four or five different tests will be necessary.

Precautions.—Before commencing a boiler test, it is advisable to make a general survey of the boiler on the day previous. Boiler flues, plates, tubes, &c., should be well cleaned inside and out, and the brickwork examined for cracks and fissures. These can be found by firing with smoky coal and then shutting the damper, when smoke will issue from them. When found they should be stopped with ganister or fire-clay. It will not be found sufficient to plaster them, but the ganister should be rammed well in, as otherwise, when dry, it will only crack off. The boiler should be worked for some time before the commencement of the test, so that the brickwork and flues are heated up to their normal condition.

Log-Sheets.—Log-sheets should be prepared for the various readings to be taken, and each assistant should be carefully instructed as to his duties. The duration of the trials should be about eight hours. Arrangements should be made for carefully weighing the coal, and it will be found convenient to put it into buckets or baskets each containing 100 lbs. The time at which each bucketful is given to the fireman should be noted, and a fresh portion should not be delivered until the previous one has all been fired. Samples of the coal used should be taken at intervals during the test, for analysis. The weight of ashes should be carefully recorded, and samples taken from it for a subsequent analysis to determine the proportion of unburnt fuel. The boiler being tested should be fed by a separate pump from that used for any other boilers that there may be in the station; and as for testing purposes it is advisable to eliminate as far as possible all sources of leakage, and therefore inaccuracies, it is important to blank-flange the feed and blow-off pipes of the boiler from other connections.

Feed-Water Measurement.—Local conditions usually determine the most convenient method of measuring the feed-water.

If approximate results, correct to within 1 or 2 per cent, are considered sufficiently accurate, a meter may be used. It should be calibrated before the test; a meter may have been constructed to measure either hot or cold water, and the difference of volume, if hot-feed is passed through a meter intended to measure cold water, may make a considerable difference.

The most accurate method is undoubtedly to measure the water in tanks; two of these should be used, one being placed above the other, so that the water contained in it may run by gravity into the lower one. Each tank should have a graduated scale in it indicating the quantity of water contained when the level is at any point. The upper tank is filled from the source of supply, and then emptied into the lower one through a valve at the bottom; the quantity of water the tank holds being known, it is only necessary to record the time at which its contents are discharged into the lower tank.

The feed-pump draws constantly from the latter, thus permitting the boiler to be fed regularly, whether the upper tank is being filled or emptied. Care must be taken that the pipe by which the upper tank is filled is of such a size that the time occupied in filling and emptying is less than the time taken by the feed-pump to empty the lower tank, otherwise the supply of water to the boiler will be intermittent. The size of the tanks should be determined by the quantity of water to be evaporated per hour in the boiler; it is very inconvenient, and causes much loss of time to be compelled to fill and empty the upper tank at too frequent intervals.

Water-Level.—The water-level during a test should be kept as constant as possible. The level at the commencement of the test should be marked on a piece of wood fixed between the gauge-glass mountings, and at the end of the test the water should be at the same level, or slightly below it. If the latter, the necessary water should be pumped into the boiler, and the quantity added to the quantities already recorded in the log-sheet as having been evaporated during the period of test.

Curve.—It will be found useful to plot a curve, as the test proceeds, of the amount of fuel and water used. If the curve shows any sudden fluctuations then possibly some error has been made, which may be rectified at once, thus preventing the results of the test being vitiated.

Condition of Fires.—The fires should be kept as low as possible at the beginning and end of the test, so that the same amount of coal may be on the grate at each period. A large fire is not required.

Readings.—Besides the measurements of coal and water, readings at regular intervals should be taken of the gauge pressure and also of the temperatures of the feed-water, gases leaving the boiler, air, and, if superheaters are used, of the steam.

Unless a thermo-electric pyrometer is used, pyrometer readings of the temperature of the flue gases are of somewhat doubtful accuracy, and more certain results can be obtained, although necessarily within fairly wide limits, by using metals which have known melting-points. Below is a table giving the temperatures at which various metals melt:—

	C. Degrees.	F. Degrees.
Tin	232	449.6
Bismuth	260	500
Lead	326	618.8
Zinc	420	788
Magnesium	500	932
Antimony	632	1169.6
Aluminium	655	1211
Brass	900	1652
Silver	960	1760
Gold	1065	1949
White cast-iron	1075 1135	1967 2075
Copper	1080.5	1976
Gray cast-iron	1200 1250	2192 2282
Steel	1375	2507
Nickel	1500	2732
Wrought-iron	1500	2732
Palladium	1580	2876
Platinum	1780	3236

Small rings of various metals may be easily obtained for the purpose of temperature observations. A set of the rings is slipped on to an iron rod, the rings being in the order of their melting-points, and the rod inserted through a hole in the flue; all those rings whose melting-points are below the temperature of the gases will disappear, and that temperature is known to be between the melting-point of the first remaining ring and the ring which was next to it.

The draught at the damper should be measured by observing the difference of water-level in a glass U-tube.

After the test has been completed, the entries in the log-sheets are collected, and the results embodied in a test-sheet. The usual form of this is given below.

....190

Test made at...

By..on...

Description of Boiler...

....

....

1.	Heating surface	sq. ft.
2.	Grate surface (ft. in. long × ft. in. wide)	sq. ft.
3.	Ratio of heating surface to grate surface	
4.	Kind of fuel used	
5.	Duration of test	
6.	Average observed gauge pressure	lbs.
7.	Average temperature of water fed to the boiler by Feed-Water Heater or Economizer	deg.
	(Average temperature of water supplied to the Economizer)	deg.
8.	Pounds of fuel fired	lbs.
9.	Pounds of ashes	lbs.
10.	Pounds of combustible (fuel-ashes)	lbs.
11.	Per cent of ashes	per cent.
12.	Coal consumed per sq. ft. of grate per hour	lbs.
13.	Total water evaporated	lbs.
14.	Water evaporated per hour	lbs.
15.	Water evaporated per sq. ft. heating surface per hour	lbs.
16.	Water evaporated per lb. of coal. Actual conditions	lbs.
17.	Water evaporated per lb. of coal, assuming feed-water at 212°, and under atmospheric pressure	lbs.
18.	Water evaporated per lb. of combustible. Actual conditions	lbs.
19.	Water evaporated per lb. of combustible, assuming feed-water at 212°, and under atmospheric pressure	lbs.
20.	Quality of steam	
21.	Temperature of boiler-room	deg.
22.	Temperature of flue gases	deg.
23.	Force of draught, in inches of water	inches.
24.	Calorific value of fuel by calorimeter	B.T.U.
25.	Calorific value; theoretical evaporation from and at 212°	per lb.
26.	Efficiency: heat utilized	per cent.

Methods of reducing the evaporation to standard conditions, *i.e.* from and at 212°, and of calculating the efficiency of the boiler, have already been explained.

Such a test as described above is sufficient in a great many cases, but where it is desirable to obtain the fullest possible information as to what the boiler is doing, or to trace the cause of any unsatisfactory results, the flue gases should be analysed at regular intervals throughout the test. This may be done easily by the aid of the Orsat apparatus, which may be used by anyone after a very small amount of practice. If a gas analysis has been made, it is possible to construct a balance-sheet of the heat supplied and used.

The following is a specimen heat balance-sheet:—

To calorific value of 1 lb. of coal ... ... B.T.U. 12,870

B.T.U. Per cent.

1. By heat absorbed by water in boiler = evaporation from and at 212° per pound of combustible × 966 ... 9177 ...... 71.30

2. By heat lost through moisture in coal

$$= \frac{M}{100}\left(1091.7 + .305\,(t_1 - 32) - t_2 - 32\right) \quad \ldots \quad \ldots \quad 25.47 \quad \ldots\ldots \quad .20$$

M = per cent moisture in coal.
t_1 = temperature flue gas.
t_2 = „ boiler-house.

3. By heat lost in ash (unburnt coal, &c.) $= \cdot K\,(A - d)$ ... 643.50 ...... 5.00

K = caloric value of coal.
A = observed ash per pound coal.
d = ash by analysis per pound coal.

4. By heat lost in chimney gases.

Convert volume per cent of flue gas into weight per cent, thus—

$$\begin{array}{rrlcrr}
\text{If } CO_2 = & 11 & \text{per cent volume} & \times \text{ density} & 22 = & 242 \\
O = & 6 & \text{„ \quad „} & \times \text{ „} & 16 = & 96 \\
N = & \underline{83} & \text{„ \quad „} & \times \text{ „} & 14 = & \underline{1162} \\
 & 100 & & & & 1500
\end{array}$$

Then weight per cent—

$$CO_2 = \frac{242 \times 100}{1500} = 16.1 \text{ per cent by weight.}$$

$$O = \frac{96 \times 100}{1500} = 6.4 \quad \text{„} \quad \text{„}$$

$$N = \frac{1162 \times 100}{1500} = \underline{77.5} \quad \text{„} \quad \text{„}$$

$$100$$

Multiply by specific heats—

$$\begin{array}{rlcr}
CO_2 = & 16.1 \text{ per cent} & \times .217 = & 3.493 \\
O = & 6.4 \quad \text{„} & \times .218 = & 1.397 \\
N = & 77.5 \quad \text{„} & \times .244 = & \underline{18.910} \\
 & & & 23.800 = M.
\end{array}$$

Weight of air used per pound carbon in 1 lb. coal = G.

$$G = \frac{N \times 4.762}{C}$$

N = per cent N by weight.
C = per cent CO_2 „

Weight of air used per pound coal = A.

$$A = \frac{G \times J}{100}$$

J = per cent carbon in coal.

Loss in waste gases—

$$= \frac{M}{100} \times (t_1 - t_2)\,A \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad 1786.00 \quad \ldots\ldots \quad 13.80$$

5. Unaccounted for ... ... 1238.03 ...... 9.70

12870 100

CHAPTER XIII

A MODERN BOILER-HOUSE

To conclude this section of the work it may be of interest to describe an up-to-date boiler-house plant, and a typical installation is that of the Pinkston station of the Glasgow Corporation Tramways.

The coal used is what is termed "washed nuts", and is a suitable coal at a comparatively cheap price for the mechanical chain-grate stokers employed. The coal is dumped into eight receiving hoppers, from whence it falls into bunkers outside the end wall of boiler-house, which have a capacity of 4000 tons. From these bunkers the coal drops to the conveyors, of which there are two, one on each side of the boiler-house. Each conveyor is 860 feet long, and has a vertical lift of 107 feet. The conveyors are driven by a shunt-wound electric motor running at 200 revolutions per minute, with an E.M.F. of 500 volts, the speed being reduced by means of spur-gearing. The coal is conveyed by the conveyor to bunkers above the boilers, a special dumping gear being used for upsetting the buckets and delivering the coal at any point. Each conveyor can handle 50 tons of coal per hour at a speed of 45 feet per minute. The overhead bunker runs down the centre of the boiler-house from end to end of the building, and the bottom of the bunker is so arranged as to form a hopper for each pair of boilers, with an opening in the bottom of 16 square inches; it is controlled by a coal-valve.

Each pair of boilers is provided with a self-contained trolley-hopper with shoot and weighing-scale running on rails, so arranged that it can be placed underneath the bottom of the valve of the overhead bunker, and by opening the valve can be filled with coal, which is weighed by the scale. The coal then passes down the shoot into the hoppers of the stokers. About 400 pounds of coal are weighed at each feed.

There are thirty-two chain-grate stokers, two to each boiler, a description of which will be found on page 48. The ashes pass over the dumping-plates of the stokers on to the ash-doors, and when the latter are opened the ashes gravitate down below the boiler into a shoot; by means of fillers they are taken to the conveyor, and so away to the ash-bunkers, whence they are either carted away, or taken away in railway trucks. The stokers are operated by underground shafting driven by two Westinghouse electric motors.

There are sixteen Babcock & Wilcox boilers. Each boiler has 5173 square feet of heating surface, the grates being 10 feet 10 inches wide × 8 feet 6 inches long, and each boiler can evaporate 20,000 pounds of water per hour at a pressure of 160 pounds per square inch. The boilers are arranged in two lines, one on each side of the boiler-house, there being altogether eight batteries of two boilers each. A general description of this type of boiler will be found on page 19.

The boilers have two steam-and-water drums, 4 feet diameter, connected by a steam cross drum 2 feet diameter; there are eighteen sections in each boiler, each section being thirteen tubes high, the tubes being 18 feet long and 4 inches diameter. Each boiler is fitted with a steam superheater.

A special feature of this installation is the overhead flue. There are two of these, constructed of steel, and they are each about 176 feet long, supported on steel girders running over the top of boilers. The flues are lined with fire-brick.

There are two fuel economizers, one for each line of boilers, each capable of dealing with 120,000 pounds of water hourly, raising the feed-water to about 160° F. The scrapers for keeping the tubes of the economizers free from soot are motor-driven.

There are two chimneys, one for each line of boilers, built of brick. The height of the chimneys from the boiler-room floor to the top of coping is 250 feet, and the diameter inside the top of chimneys is 16 feet 9 inches. The chimneys are lined with fire-brick, and the top is provided with a cast-iron cap, made in sections and bolted together.

Factory Act.—In conclusion, it is thought advisable to quote from section 2 of the Factory and Workshops Act, 1901, which came into operation on 1st January, 1902, and which provides that every occupier of a factory, or workshop, or any other place to which the Act applies, must have his boilers examined at least every fourteen months by a competent inspector, for whose competence he will be held responsible.

1. Every Steam-Boiler used for generating steam in a factory or workshop, or any place to which any of the provisions of this Act apply, must, whether separate or one in a range—

(*a*) have attached to it a proper safety-valve, and a proper steam-gauge and water-gauge, to show the pressure of steam, and the height of water in the boiler.
(*b*) be examined by a competent person at least once in every fourteen months.

2. Every such boiler, safety-valve, steam-gauge, and water-gauge must be maintained in proper condition.
3. A report of the result of every such examination in the prescribed form, containing the prescribed particulars, shall within fourteen days be entered into, or attached to, the General Register of the Factory or Workshop, and the report shall be signed by the person making such examination, and if the person is an Inspector of a Boiler-inspecting Company or Association, by the Chief Engineer of the Company or Association.
4. A factory or workshop in which there is contravention of this Section shall be deemed not to be kept in conformity with this Act.

2. Steam-Engines and Other Prime Movers

Introduction.—It is not the intention in this article to go through the elementary theory of the heat-engine, but rather to draw attention to the salient features of engines designed for the express purpose of driving electric generators. It is assumed, therefore, that the readers of this work are acquainted with the fundamental principles underlying the design of heat-engines, and it is intended to discuss only those qualities which engines must possess, if they are satisfactory to drive dynamos. The order in which these necessary qualities will be discussed is:—

1st. The "Power and Efficiency" of engines when put to drive generators for particular service.
2nd. The degree of steadiness of turning which is necessary, coming under the heads of "Governing" and "Fly-Wheels".
3rd. Economy of steam or gas, coming under the head of "Consumption".

Each of the above qualities will be treated of in three divisions—Reciprocating Steam-Engines, Steam-Turbines, and Gas-Engines.

CHAPTER I

RECIPROCATING STEAM-ENGINES—POWER AND EFFICIENCY

Power of Engines for Electrodeposition of Metals.—The demand made on the electric generator, when once the process of electrodeposition has been started, is that it should supply energy at a constant (or nearly constant) rate for long periods at a stretch. This being so, the engine should be of such a size that it will exert sufficient power to drive the dynamo at the required output. No reserve of power is required beyond, perhaps, a very small margin to guard against any temporary falling-off in efficiency of the set, due to lack of regular attention to his duties on the part of the driver.

To take an example, assume that the output of the dynamo is to be 300 kilowatts, or 402.1 electrical horse-power. The generator would certainly be designed so as to be very efficient at this output, and it is not too much to assume that it would have an efficiency of 94 per cent at least. Under these circumstances the power necessary to drive the dynamo will be 402.1 ÷ 0.94 = 427.8 horse-power. This will be the effective or brake horse-power required from the engine, provided the armature of the dynamo is coupled direct to the crank-shaft of the engine. Belt-driving of dynamos is rapidly being given up, on account of the loss incurred in merely running a belt, and will soon be a thing of the past.

In high-class engines the mechanical efficiency, or ratio of brake horse-power to indicated horse-power or power generated, varies from 89 to 93 per cent, according to the condition and excellence of design of the engine. Taking 91 per cent as a fair average figure, it is seen that the engine, to drive the 300-kilowatt dynamo under discussion, must have its cylinders so proportioned that the steam generates 427.8 ÷ 0.91 = 470.1 horse-power in the cylinders. This will be the indicated horse-power (I.H.P.) the engine *must* have. For the reason already given, engine builders would, in all probability, reckon that at times 490 I.H.P. might be required, and would provide accordingly. Further, it may be mentioned that the engine would be so proportioned that it would work with the greatest degree of steam economy at this output of 300 kilowatts. The subject of steam economy, however, is one that will be discussed later on.

Power of Engines for Electric Lighting.—An engine designed to drive a generator solely for electric-lighting purposes needs to have much the same margin of power as the engine just considered, but the conditions of working are rather different. The output will, in all probability, vary, gradually increasing from the time of starting up the engine till the time of maximum demand for light. Afterwards the load will gradually decrease, until there is no longer sufficient load to warrant the engine being run at all. Obviously there is not much more to be said, with regard to maximum power, than has already been said in connection with the depositing dynamo set. The chief points of difference between the two sets are matters of steam economy and of governing rather than of power.

Power of Engines for Electric-Traction Work.—The power which ought to be provided for driving a traction generator is a more difficult matter to determine, because the demand made on the generator for current is a constantly varying one. Engineers, asked to prepare a scheme of electric traction for a town, have to determine the number of tram-cars which are to be in service as a rule. They have also to estimate how many of these tram-cars will be running on an up-grade, how many on a down-grade, how many starting and how many standing, at any moment, and generally, to come to a conclusion as to what will be the normal demand for current from the generating station.

Having settled on this normal demand, and fixed upon the number of generators it is advisable to have in the generating station, to guard against stoppage of the service through accident or through engines undergoing repair, it is necessary to consider what power should be provided in each generating set to meet temporary demands for current above the normal. This varies very much according to the nature of the route, whether the habits of the population increase the amount of traffic at stated hours of the day, and so forth. Further, at times there may be many cars starting at the same moment and calling for a large amount of current to get up speed. At other times many cars may be standing, and therefore using no current at all. The demand on the generators for current is, therefore, often a very varying one, being perhaps 30 per cent above the normal one minute and perhaps 30 per cent below the normal the next minute. Generally speaking, the traction generator should be able to supply current over and above the normal by 10 to 15 per cent for an hour at a time, 20 to 30 per cent for a few minutes at a time, and even as much as 45 to 50 per cent momentarily, though this latter margin of power would be needed more to guard against total stoppage of the generator through a short circuit than for any other reason.

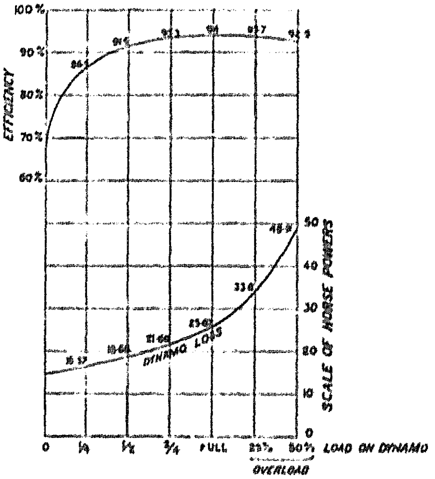


Fig. 1283.—Dynamo Efficiency

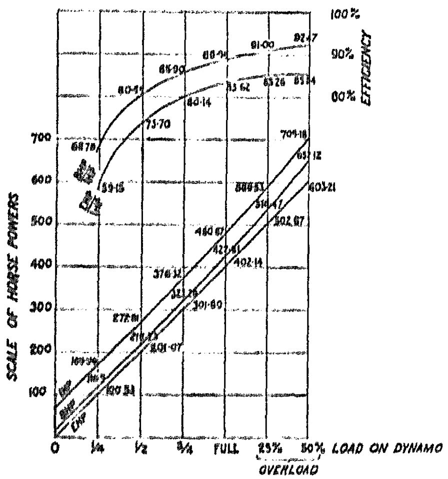


Fig. 1284.—Engine Efficiency and Efficiency of Combined Engine and Dynamo

Assume now that the unit selected for a certain traction generating station is a 300-kilowatt set (normal output) as before. The efficiency of a traction generator, having 94 per cent as its efficiency at normal full load, will vary somewhat according to the curve given in fig. 1283. Having given the dynamo efficiency at all loads as in fig. 1283, it is next possible to set down another diagram which will show the horse-powers required to drive the dynamo. Taking as base the load on the generator expressed as percentages of the normal output, first draw the straight line representing the electrical horse-power (E.H.P.) corresponding to the kilowatts output of the dynamo (fig. 1284). On this E.H.P. line plot the loss in horse-power due to friction of the dynamo bearings and brushes, and electrical losses in the dynamo at any number of loads thought desirable. These losses may be taken from the lower curve in fig. 1283. Draw a fair curve through the points so plotted, and the brake horse-power

(B.H.P.) line is obtained. This line represents the power that must be applied to the generator shaft to drive it.

The next step is to draw a line parallel to the B.H.P. line in fig. 1284, but above it by the amount of the frictional losses in the engine. This line is the indicated horse-power (I.H.P.) required in the cylinders of the engine. As to the "amount" of the frictional loss in the engine, it may be said that, as the 50-per-cent overload is only required for a moment now and again, this load may be considered only in connection with the "strength" of the parts of the engine, but without particular regard to the "wear" of the parts. The 25-per-cent overload, however, being required for an hour at a stretch two or three times a day, must be allowed for almost as amply, as regards bearing surfaces, as if it was the normal full load of the electric-lighting engine already spoken of. Consequently the efficiency of the traction engine at 25 per cent overload will be practically identical with that of the electric-light engine at full load, viz. 91 per cent in the case under consideration. The I.H.P. line, therefore, will lie, to scale, $536.5 \div 0.91 - 536.5 = 53$ horse-power above the B.H.P. line at all points of the diagram. The figure may be completed by drawing the efficiency curves of the engine (B.H.P. ÷ I.H.P.) and of the combination (E.H.P. ÷ I.H.P.), and all particulars as to horse-power and efficiency are visible at a glance.

It is seen, from fig. 1284, that the engine to drive this 300-kilowatt traction generator must indicate—

Normally,	480 I.H.P.				
Sometimes,	590 „	for 25	per cent	overload.	
Rarely,	705 „	„ 50	„	„	

Figs. 1283 and 1284 may be taken as fairly representative of the efficiency obtainable from a good dynamo coupled *direct* to any good class of reciprocating steam-engines.

Driving the dynamo by belt can hardly be called modern practice. It introduces a loss which may easily amount to 5 or even to 10 per cent of the total horse-power of the engine, without any compensating advantages.

CHAPTER II

RECIPROCATING STEAM-ENGINES—GOVERNING

The parts of an engine which have the power to affect the steadiness of running are the governor, and those rotating parts which perform the functions of fly-wheels.

The duty of the *governor* is to permit the passage of just so much steam into the cylinders as will serve to drive the engine at the intended number of revolutions per minute, no matter what load the engine may be working against. The governor has to maintain the speed of turning at a certain predetermined and approximately constant number of *revolutions per minute.*

Fly-wheels, and those other parts which by reason of their rotation have the same effect as fly-wheels, have no power to influence the number of revolutions per minute made by the engine, but have a very great effect upon the "cyclic regularity" of turning of all reciprocating engines. Anybody who has seen a hydraulic pump of the fly-wheel type at work has noticed how the fly-wheel will at certain parts of the revolution be moving slowly, and at other parts of the same revolution be moving quickly. Although this hydraulic pump may be making its regular 20 or 30 revolutions per minute, minute after minute, it has a very great "cyclic irregularity" of turning. No reciprocating engine turns with an absolutely regular angular velocity, there are always points of maximum and of minimum velocity each revolution. The "cyclic irregularity" of turning is usually expressed as a fraction having the difference between maximum and minimum velocity as numerator and the mean velocity of rotation as denominator.

$$\text{Fluctuation of velocity} = q = \frac{V_H - V_L}{V_m}.$$

Where V_H is highest, V_L is lowest, and V_m the mean velocity.

The governors chiefly in use at the present day are the spring-loaded centrifugal governor and the shaft governor which makes use of the inertia of its weights as well as of their centrifugal force.

Simple Centrifugal Governors are either carried by the engine shaft or are driven by it through some form of gearing. The balls are carried on arms pivoted on a cross-head rotating with the governor spindle, and are so placed that the path of the balls as they fly out or in is in the same plane with the axis of the governor spindle. The only force utilized is the centrifugal force of the balls. These governors are used to control the position of some small valve such as the throttle-valve of the engine, or the valve admitting steam to a relay cylinder, or to regulate the position of some detaching lever where trip-gear on Corliss or drop valves is in use. They are only used, in fact, when no great amount of power is required of them.

Shaft Governors are always carried direct on the engine shaft. The weights or balls are pivoted on pins fixed in a disc keyed on to the shaft, and are so placed that the path of the balls as they fly out is in a plane at right angles to the axis of the shaft. Owing to this arrangement it is possible to utilize the inertia force of the weights to assist their centrifugal force. These governors are used where great power is required, the weights being attached to the pins or eccentrics which drive the distributing valves of the engine.

Methods of Governing.—There are three methods of governing in vogue at the present day—throttle governing, expansion governing, and a combination of both throttle and expansion governing. By the first method the *pressure,* and therefore the density, of the steam taken into the cylinder each stroke, is varied according as more or less work is required from the engine. The steam in flowing from the boiler to the engine passes through a *throttle-valve*, which is open to a greater or less extent according to the speed at which the engine is running, the valve being under the control of the governor. By the second method the *volume* of steam taken into the cylinder is varied, the pressure remaining constant. This is effected by means of altering the "cut-off" of the valves to suit the demand for power. There are several ways of doing this automatically by means of the governor, descriptions of which follow. The third method was devised with the view of improving the steam economy of an engine at very light loads. A special way of doing this is adopted in the case of the Willans engine, and is described later.

The function of the governor is to perform any one of these operations automatically as the speed of the engine tends to increase or decrease, and so to keep the engine turning at as nearly as possible a constant number of revolutions per minute. Governors also have an attachment whereby the normal speed of rotation of the engine may be varied by hand while the engine is running. The range of speed variation so obtainable is often as much as 25 per cent, though more commonly 5 to 10 per cent is found to be sufficient in practice.

Governing Engines for Electrodeposition of Metals.—As the load on such engines is constant, or very nearly so, no very great refinement in governing is required. The engines should be proportioned to give the best economical result possible at the normal full load, and the points of cut-off adjusted once for all to ensure such results. The best engines for such work are triple-expansion or quadruple-expansion condensing engines fitted with simple centrifugal governors operating throttle-valves. There is no use for elaborate valve gears and governors, seeing that, the load being constant, there is no work for them to do. The extra friction of automatic expansion gear would only diminish the economy in steam of the engine, besides adding needlessly to the prime cost of the engines.

Governing Engines for Electric Lighting.—These are engines which need to have fairly sensitive and, more important still, very stable governors. The load on the engines being a gradually varying one, the governors should be sensitive enough to feel slight increases or decreases of speed and stable enough to control them without showing any tendency to "hunt". An accuracy of governing of from 3 to 5 per cent will be close enough for all practical purposes; this means that a 300-kilowatt electric-lighting set running at, say, 300 revolutions per minute under full load, should not, when the dynamo is excited but delivering no current, exceed 309 to 315 revolutions per minute.

As far as speed regulation is concerned, the methods of throttle or expansion governing are equally applicable to this engine, but the question
ing will best suit the needs of the particular town
amo is to light will turn upon the nature of the
plant. If the engine can be run up fairly quickly
full load till shortly before it can be stopped
he throttling method is the better on account of

its simplicity, coupled to the fact that a throttle-governed engine gives its best economy at full load. If, however, the engine has to be run for long periods at about five-eighths to three-quarters of full load it is just a question whether expansion governing may not be more suitable, because it is possible to arrange an expansion-governed engine to give its best economy at five-eighths to three-quarters load. There is, however, no very marked advantage even in this respect, taking into account the extra cost of the automatic expansion gear over that of the more simple throttle-valve gear.

A method often adopted is to fit the engine with a throttle governor, and expansion gear which allows the cut-off to be altered by hand as the load gradually changes. The effect of the two methods of governing on economy of steam consumption will be discussed later on.

Governing an Electric-Traction Engine.—The governing of an electric-traction set is a more difficult matter than that of either of the two previous cases. The demand for current made on the generator is incessantly varying, unless the generator is run in parallel with an accumulator. Assuming that the generator is run direct on the feeders, and no accumulator used, the governor may no sooner have settled the engine down to run steadily at one rate of supply of current than the demand alters suddenly and possibly largely. On change of load, the engine will try to race or slow down according to whether load is taken off the dynamo or put on, and the governor balls take up a new position, thus admitting less or more weight of steam to the cylinders as required. These changes may occur so rapidly and violently that it may be said that the governor does not get a chance to settle the set down to any one particular rate of output, before it has to respond to a new condition of things. Under these circumstances a very quick-acting governor is required.

Here, again, both the throttle and the expansion methods of governing are in successful and satisfactory use as far as controlling speed is concerned. In the case of the throttle-governed engine, however, a heavier fly-wheel is required than in the case of the expansion-governed engine. The throttle-valve controls the supply of steam passing *into* the engine, but cannot control the action of the steam already in the cylinders and receivers. Conceive the throttle-governed engine to be running under full load and suddenly a large amount of the load to be removed. The engine speed will begin to rise, and the governor will immediately shut off steam; but there is already a full charge of steam in the engine receivers and passages, which charge is beyond the control of the governor. This steam will give up its energy to the pistons although it is not required, and the speed of the set will increase still more in consequence, until the supply of steam being admitted to the engine has just less than sufficient energy to do the work of driving the generator at the new output. From this time the fly-wheel will help the steam supply and gradually lose energy, and therefore velocity, until a speed is reached at which the throttle-valve is just open wide enough to allow sufficient steam to pass to drive the set against the load without any help from the fly-wheel; thus the normal speed is again reached and maintained until some further change of load occurs. This *momentary change of speed* on change of load has to be kept within bounds by having a fly-wheel such that the energy in the steam already past the throttle-valve when it closes may not increase the speed of rotation by more than a certain allowable percentage. The momentary change of speed when full load is suddenly removed is usually kept down to 5 or 6 per cent of the normal speed by means of a suitable fly-wheel.

In the case of automatic expansion governing, where the cut-off *in all cylinders* is under control of the governor, a change in speed alters the weight of *each* cylinder charge of steam, and the only steam beyond the control of the governor is that already in the cylinders, which is exhausted immediately on the return stroke. It will be recognized, therefore, that automatic expansion governing has advantages in controlling the speed of a traction set not possessed by throttle governing. Those engines in which the cut-off is varied only on the first cylinder, take up an intermediate position between simple throttle-governed engines and complete expansion-controlled engines.

There is yet one other method of governing a traction set, sometimes provided in conjunction with the throttle-valve method. It consists in providing a small supplementary valve under control of the governor, which admits high-pressure steam from the main into the low-pressure steam-chest when the speed has sunk slightly below the normal speed for full load. This method is extremely useful, because it enables the cut-off to be fixed at an earlier point of the stroke than could otherwise be done. In the case of 25 per cent overload being required only momentarily, it is obvious that so long as it is obtained it does not matter at what cost of steam. The cut-off, then, when the supplementary valve is fitted, may be fixed so that the normal load can be just obtained with the full steam pressure available, and any overload will be met by the supplementary valve opening and allowing extra steam to pass into the low-pressure steam-chest.

A further advantage this method possesses is, that upon the sudden imposition of a big load the momentary fall in speed of the engine will cause the supplementary valve to open, and so the load will be picked up more quickly than if the extra steam had to find its way into the engine by the ordinary path through the high-pressure valves and cylinder.

Centrifugal Governor for Throttle Governing.—Fig. 1285 is a good example of the centrifugal governor coupled to a throttle-valve, and illustrates the Paxman "Peache" governor as applied to high-speed engines. The governor balls *a* are shown in the position they would occupy if the engine were running "light". The throttle-valve *b* is as nearly as possible closed, and the steam passes through from the steam main to the engine steam-chest as indicated by the arrows. The governor balls *a* are carried in brass bushes *c* fixed in the two arms of a revolving cross-head *d* driven by a shackle *f*, which in turn is driven by the pulley *p*. The balls or weights are of peculiar shape, and consist of a spherical ball proper *a* on an arm carrying a counterbalance weight *w* on the opposite side of the fulcrum *c*.

To reduce friction in a spring centrifugal governor it is desirable that the centrifugal force of the weights, and the centripetal action of the spring or springs, shall be directly opposed to each other without the intervention of joints or levers. In most spring governors the balls are directly tied together by helical springs, but although by this means friction due to the interaction between the balls and springs is reduced to a minimum, yet there is difficulty in obtaining close regulation of speed with such governors. The reason for this is, that since the centrifugal force of the balls or weights (the speed being constant) varies directly as the distance of their centres of gravity from the axis of the governor shaft, the centrifugal force would not become "nil" until the centres of gravity of the balls coincided with the axis of the governor shaft. The tension of a helical spring varies as its length, consequently, if the points corresponding with the centres of gravity of the balls are to be directly tied together by helical springs, it is obvious that an isochronous governor cannot be made unless the springs have no length when at rest, which of course is impossible.

The balance-weight *w* is of such a size that the

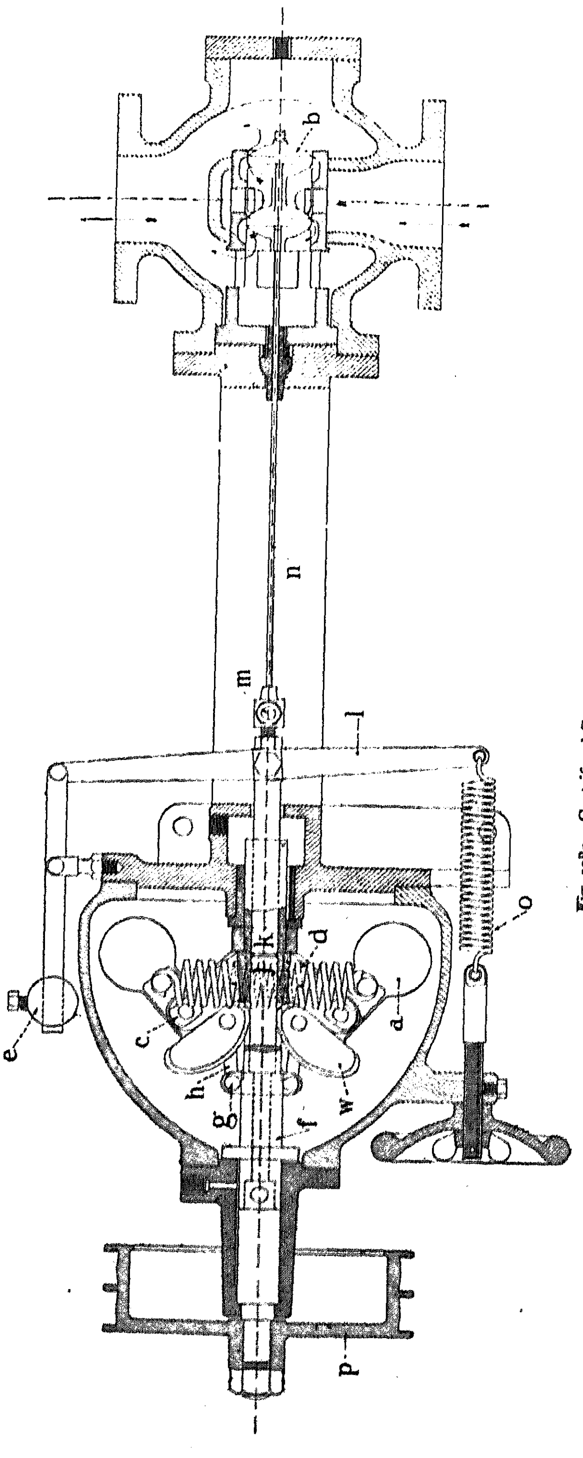


Fig. 1285.—Centrifugal Governor. Peache Type

net movement of the centrifugal forces of weight *w*, arm, and ball *a*, *about the pivot c*, becomes "nil" at some point outside the axis of the governor shaft, and thus it is possible to employ a spring of practical dimensions which will at all positions exactly balance such movement. By the employment of the balance-weight *w* it is further possible to directly balance the centrifugal force of the balls and balance-weights by the centripetal action of the springs, and for this purpose the points of spring attachment are located at such a position of the governor arms that, in the mean working position, the movement of the centrifugal forces of the ball and balance-weights balance each other *round the point of spring attachment.* The balance-weights are slotted and drilled to receive the links *h*, making connection with the cross-head *g* on the sliding spindle *j*, which slides inside the shackle *f* and the cross-head *d*, and rotates with them. Another spindle *k*, also capable of sliding in the governor cross-head, but prevented from turning by the set pins in the lever *l*, carries a clip-piece *m* into which the throttle-valve spindle *n* is fastened. The spindle *k* is always kept in contact with the spindle *j* by means of the tension of the "speeding" spring *o*.

The action of the governor is as follows:—

At a certain speed the centrifugal force of the balls is sufficient to enable them to take up an extreme position (shown in fig. 1285) against the pull of the two governor springs, directly connected from arm to arm, and against the pull of the spring *o* transmitted through the lever *l*, spindles *k* and *j*, and links *h*, on to the balls. Upon the engine slowing down, these spring tensions are enough to overpower the centrifugal force of the weights, and to pull them in, at the same time pulling the throttle-valve *b* to the left, and so increasing the distance between the cut-off edges of the valves and the edges of the valve seat, to allow the passage of more steam to the engine.

For effecting changes of speed an adjustable spring *o* is employed, and transmits its force to the governor arms through leverages which vary with the position of the balls in such a proportion that the sum of the pull of main governor springs, and of the adjusting spring, against the net centrifugal force of the balls and balance-weights, for the inner and outer positions of the governor arms respectively, bear a constant ratio to each other throughout the range of the adjusting spring. In other words, the percentage of variation of speed between that corresponding to the inner and outer positions of the governor arms is kept constant throughout the range of speed adjustment effected by the spring *o*. Governing within 2 per cent from full load to no load is easy to obtain. The desired amount of error to give stability to the governor is provided in the spring *o*.

For other centrifugal governors operating throttle-valves, readers are referred to the sectional drawings of the Willans, Belliss, and Mather and Platt engines in figs. 1330, 1332, 1333, and 1334.

Centrifugal Governors for Automatic Expansion Governing.—Figs. 1286 and 1287 show the detailed arrangement of a simple centrifugal governor controlling the point of cut-off in a drop-valve engine, the drawing having been prepared from one of Messrs. Davey, Paxman, and Co.'s horizontal slow-speed drop-valve engines. The eccentrics are keyed on to a side shaft driven from the main engine shaft by bevel gearing. The top ends of the eccentric-rods are guided by side links pivoted on a pin which also carries the steam-valve lifting lever. On the eccentric-rod above the lever pin there is a catch-piece, fitted with a steel pallet, which hangs between the jaws of the eccentric-rod. As the eccentric passes its highest point the catch-piece swings over the steel pallet fitted on the end of the lifting lever, and on the downward movement of the rod engages with the lifting lever and so raises the steam-valve from its seat. The downward movement of the interlocked pallets continues until the projection on the under side of the horizontal arm of the catch-piece rides up on to the detaching lever, and so disengages the two pallets. As soon as the pallets have been drawn apart, the steam-valve falls by its own weight, assisted by the spring shown in the dash-pot, but seats itself quietly by reason of the air cushion below the dash-pot piston.

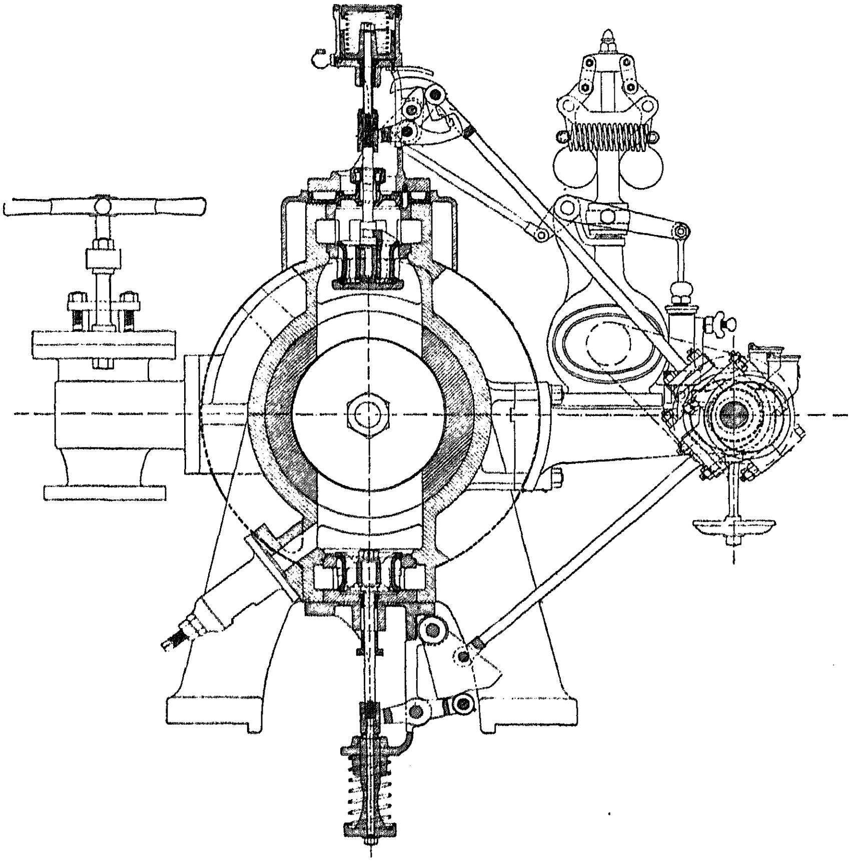


Fig. 1286.—Centrifugal Governor controlling Cut-off Valves

A very sensitive spring centrifugal governor driven from the side shaft is connected by a link to the lower end of the detaching lever. As the governor balls raise and lower the sleeve, the detaching lever is swung about its pivot, and so the catch-piece rides up on to it earlier or later in the stroke, and allows the valve to drop and cut off steam earlier or later. The gear is very simple and effective, and the quick closing of the valve gives a sharp cut-off. The valves are placed as near the ends of the cylinder as possible, thus cutting down clearance to a minimum, and also keeping initial surfaces very small. As will be explained later on, these two points are of very great importance when great economy of steam is desired. The exhaust-valves at the bottom of the cylinder are operated by eccentrics, but are not under control of the governor. The method of opening these is clearly shown in the figure, and needs no explanation.

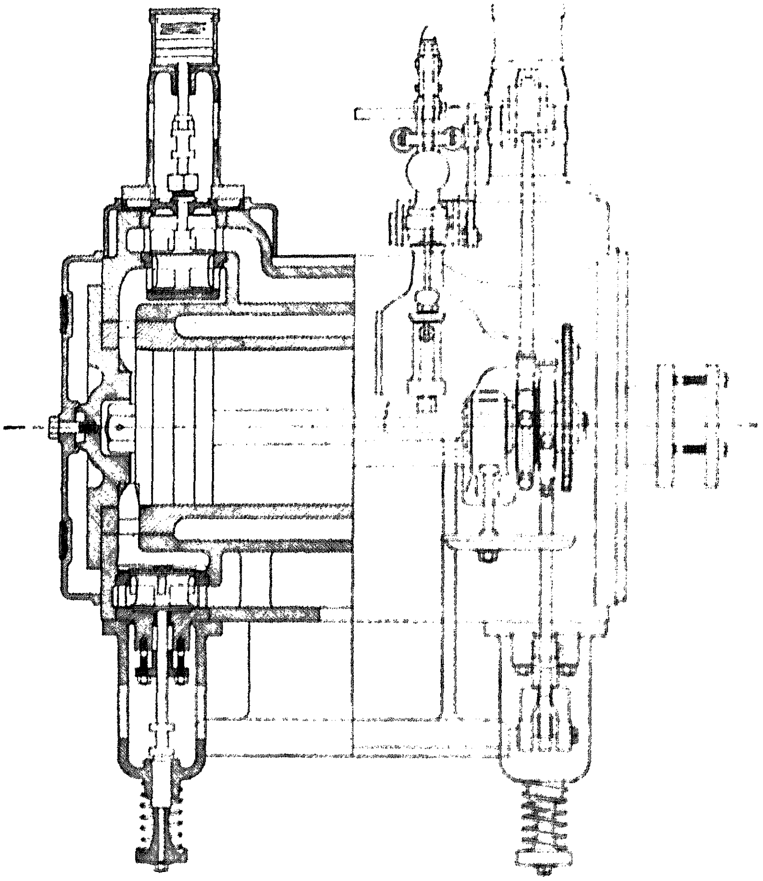


Fig. 1287.—Centrifugal Governor controlling Cut-off Valves

Shaft Governor for Automatic Expansion Governing.—Fig. 1288 illustrates a shaft governor as sometimes fitted by Messrs. Mather and Platt to their inverted vertical engines. Two weights are pivoted on a disc, which is securely keyed on the engine shaft. Those weights fly out under the action of centrifugal force against the compression of the two helical springs shown. On the side of the disc opposite to the weights they are connected across, one to the other, by a link which carries a pin working in a square bush. The bush fits between two horn-plates on an eccentric, which is free to swing about a pin fixed in the disc, as shown at the bottom of fig. 1288. As the weights fly out under the action of centrifugal force the eccentric is pulled across from the position shown in the figure, as clearly shown by the dotted centre-line through the centres of the eccentric and the pivot pin. By this means not only the throw but the angular advance of the eccentric is altered. The eccentric-rod is connected to the piston-valve spindle, and thus the cut-off and points of release and compression of the high-pressure valve are directly under control of the governor.

The chief force used by this governor is the centrifugal force of the weights, but they are so arranged that when the governor, rotating in the direction shown by the arrow, receives a sudden impulse the inertia of the weights tends to make them fly out, similarly a sudden retardation tends to make them come in towards the shaft. Thus this governor combines the principles of the inertia governor and the centrifugal governor. The centrifugal forces are, however, largely in excess of the inertia forces.

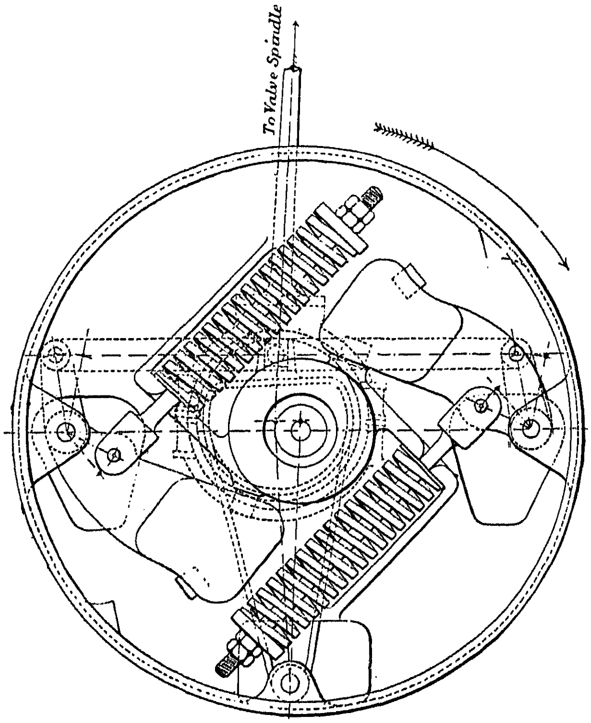


Fig. 1288.—Shaft Governor

Similar governors are used by many makers both of high-speed engines and slow-speed engines. They are used where considerable power is required for driving the steam-distributing valves of engines.

Combination of Throttle and Expansion Governing.—Fig. 1289 shows an elevation of the variable expansion gear applied to the Willans central-valve engine. Inspection of the sectional view of the Willans engine, to be found in fig. 1330, will make the method of cut-off in their throttle-governed engine clear, as well as the action of the throttle governor, which is of the usual centrifugal spring-controlled type. In the variable expansion gear the cut-off arrangements are somewhat different from those of the fixed expansion engine there described. The ports in the uppermost trunk are cut obliquely, and the trunk itself moves up and down in a hollow sleeve suspended from the top of the steam-chest. In this sleeve corresponding oblique ports are cut, and when the ports in the trunk pass below the lower edge of the ports in the sleeve, steam is cut off in the same way as when they enter the gland in the cylinder head of the fixed expansion engine. The sleeve is suspended from the rod *t*; it is fixed as regards up-and-down motion, but can be rotated upon its axis by means of the links and levers shown, either by the hand-lever *l*, or automatically by means of the steam cylinder *h*, the action of which is controlled by the governor, as explained below. Since the ports in the sleeve are cut obliquely, it follows that the ports in the trunk, when descending, meet their lower edge earlier or later in the stroke according to the relative angular position of the sleeve. Arrangements are made by which the cut-off is prevented from becoming earlier than the point where the reduction of load by throttling the steam is more economical than greater expansion. For loads less than this the engine is governed purely by the

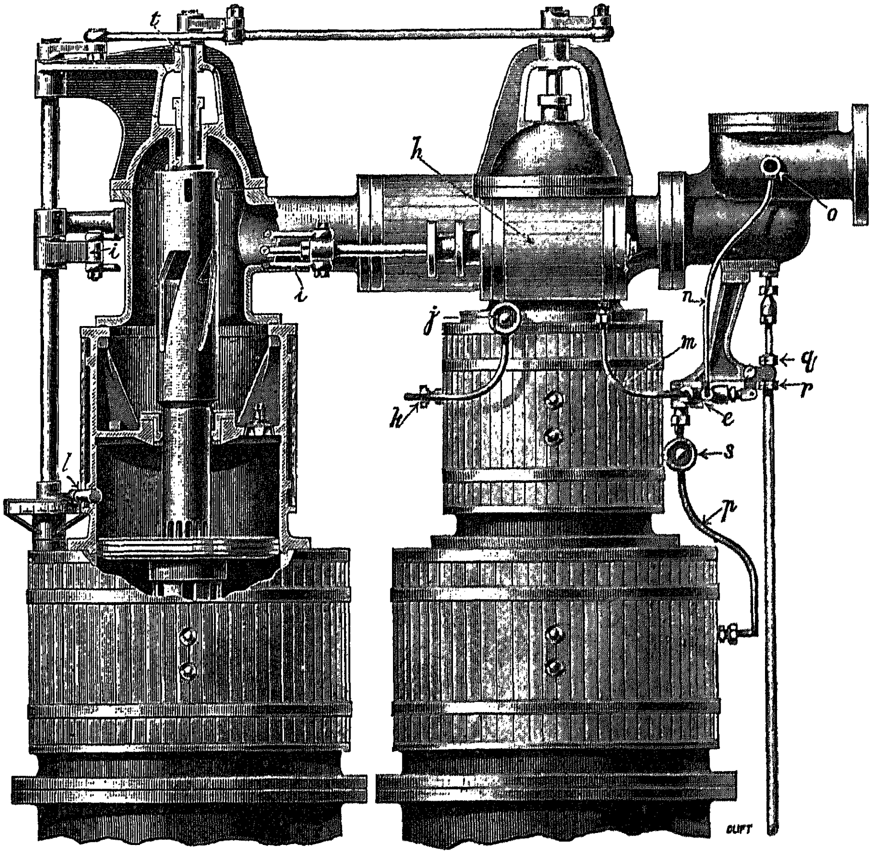


Fig. 1289.—Willans Combined Auto and Throttle Governor

throttle-valve, the cut-off for the time being fixed. For heavier loads the cut-off varies with the load, a result which is obtained as follows:—

The system of levers which moves the cut-off sleeve is connected with the piston of the relay cylinder *h* by the links and lever *i*, and the ports in the sleeve and trunk are so arranged that cut-off is made earlier when the piston moves to the left, and later when it moves to the right. The left-hand end of the cylinder is connected to the steam-chest by the valve *j* and pipe *k*, and the right-hand end is connected to a small slide-valve *e*, called the controlling valve, by the pipe *m*. This valve, by its movement, connects the right-hand end of the cylinder either to the steam-pipe outside the throttle-valve by the pipe *n* and valve *o*, or to the receiver or exhaust by the pipe *p*, but not to both at the same time. The valve is worked by a small bell-crank moved by lock-nuts *q* and *r* placed on the governor rod. When adjusted, the controlling valve is just on the point of opening when the throttle-valve is nearly full open, so that at or below the full open position of the throttle-valve the cylinder *h* is open to the steam-pipe; but if the throttle-valve rises above its full open position the relay cylinder is opened to the receiver or exhaust, and the connection to the steam-pipe is closed. In normal running the right-hand end of the cylinder is thus connected to the steam-pipe; in fact, the two ends are in communication with the two sides of the throttle-valve.

The action of the gear is as follows:—(1) If even a small portion of the load is removed from the engine the speed momentarily increases, the throttle-valve closes, and the pressure in the steam-chest, and consequently in the left-hand end of the relay cylinder, falls, while that in the right-hand end is unchanged. Under this difference of pressure the piston moves to the left, making the cut-off earlier, until about 5 lbs. below separator pressure is re-established in the steam-chest. (2) If load is put on the engine the speed momentarily drops, the throttle-valve rises, the pressure on the nuts *q* on the controlling-valve bell-crank is relieved, and the valve opens communication between the right-hand end of the relay cylinder and the exhaust, at the same time closing the steam supply through the pipe *n*; the pressure falls very rapidly in the right-hand end of the relay cylinder, and the piston moves to the right, making the cut-off later. As the pressure in the exhaust is considerably lower than that in the steam-chest, the cut-off is altered with great rapidity. When exceedingly rapid changes are needed, so as to deal with very large and sudden increases of load, the pipe from the valve may be connected to the exhaust; when such quick alterations are not required, it is often sufficient to lead it into the receiver. In the reverse action of making the cut-off earlier the difference of pressure is not usually so great, and the movement is somewhat slower; but this does not matter, as the throttle-valve holds the engine completely under control. The rapidity of this action also can be controlled by the valve *j*.

Supplementary By-Pass Valve.—It was mentioned how the cut-off in a fixed expansion engine might be adjusted to give full normal load with the full available pressure in the steam-chest, the *overload* being obtained by means of a supplementary valve under control of the governor, which allows high-pressure steam to pass into the low-pressure steam-chest. Fig. 1290 shows the arrangement of supplementary valve adopted by Messrs. Davey, Paxman, & Co.

At the end of the throttle-valve case, away from the governor, there is placed a gun-metal casting containing a miniature throttle-valve V resting in a seating *r*, and kept pressed against the main engine throttle-valve by the spring S. In the figure the main throttle-valve is shown in the position it would occupy at the maximum overload, the supplementary valve V being then open to its full extent, and the steam passing, as shown by the arrows, from the high-pressure steam-chest side of the main valve past the supplementary valve along the passage *q* into the low-pressure steam-chest. It is unnecessary to follow the action of this valve further, as this will be readily understood from the figure; it may be remarked, however, that the main throttle-valve in this position is open wider than is necessary to pass the amount of steam required for normal full load, the opening for full normal load being just when the valve V has closed the passage to the low-pressure steam-chest.

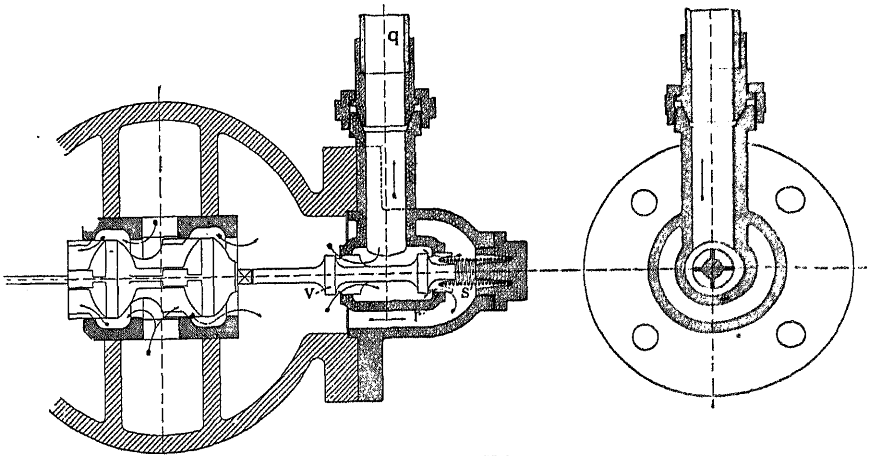


Fig. 1290.—By-Pass Valve

CHAPTER III

RECIPROCATING STEAM-ENGINES—FLY-WHEELS

For driving alternators in parallel and for polyphase machinery it is of very great importance that the "cyclic irregularity" of turning shall be very small indeed. Electricians now demand that the irregularity of turning of the engines driving their generators shall not exceed $\frac{1}{250}$ to $\frac{1}{500}$, according to the number of poles in their machines and the nature of the work to be done. Engine builders have on this account to fix upon the amount of energy it is necessary to give to the rotating parts of the combined engine and generator, in order that the allowed irregularity of turning may not be exceeded.

It has already been stated that the "governor" controls the rate of turning as regards revolutions per minute, but it has no power to affect the cyclic regularity of turning except in those cases where a shaft governor is connected direct to the slide-valve of the engine. In order that an engine may turn with absolutely even angular velocity throughout a revolution, it is obviously necessary that the resistance to turning offered by the load and the tangential turning effort exerted on the crank pin or pins of the engine shall be exactly equal for every position of the crank during the revolution. If at any position of the crank the turning effort of the engine exceeds the resistance offered by the driven machine, the speed of the combined plant will increase; for the excess of energy exerted by the engine goes to increasing the kinetic energy of the moving parts. As the mass of the moving parts does not vary, this storage of energy manifests itself in increase of speed. Conversely, there is diminution of speed when the resistance exceeds the turning effort.

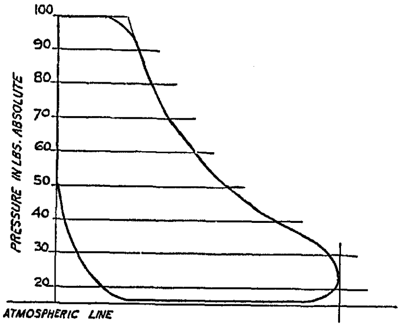


Fig. 1291

In all reciprocating engines there is such a storage and restorage of energy in and by the rotating parts, due to the unevenness of the turning effort of the engine. The successful running of alternators in parallel, and of synchronizing machinery, demands that this storage and restorage of energy shall not upset the evenness of turning by more than a certain given amount. The fly-wheel, and other rotating masses which have fly-wheel effect, are most potent in maintaining the desired approximation to absolute evenness of turning. It may be considered that any engine may, by being provided with sufficiency of fly-wheel effect, fulfil the requirements for alternator driving; but heavy fly-wheels are at best an unscientific means of obtaining evenness of rotation, and if too heavy may actually cause electrical troubles.

Crank Effort and Fluctuation of Energy.—Before it is possible to determine the fly-wheel effect necessary to secure the desired evenness of turning, it is necessary to find out the actual amount of tangential pressure at every point of the revolution on the crank-pin, or pins, if the engine has more than one crank. For simplicity of explanation, and fairness of comparison of the energy required in the fly-wheels of the different types of engine in general use, calculations have been made for engines in which the steam is used in one cylinder only before being sent to exhaust. It is not necessary to explain the way to obtain an indicator diagram, or to explain the meaning of the card drawn by the indicator, since all this has been done in great detail by the numerous writers on the steam-engine.

Taking, therefore, the case of a simple expansion engine working with 100 lbs. absolute initial pressure in the cylinder, having a "cut-off" at quarter the stroke, and exhausting to atmosphere, the indicator diagram may be somewhat as shown in fig. 1291. From this diagram the pressures acting on the piston at every point of the revolution can be measured and plotted on a base representing one complete revolution. To do this, it is necessary to erect ordinates on the diagram to represent the position of the piston at a certain number of positions of the crank in the revolution. Every 10° will be sufficient to obtain a fair degree of accuracy.

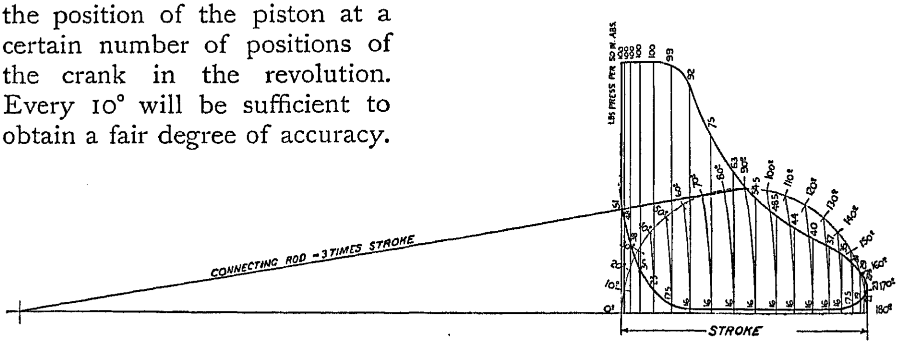


Fig. 1292

Assume, in all cases taken, that the length of the connecting-rod is six times the length of the crank. On the indicator diagram describe a circle having the length of the diagram as diameter, and divide the circumference of the circle into 36 parts. For the diagram taken from the side of the piston *away* from the crank-shaft number these divisions 10°, 20°, 30°, 40°...... beginning from the admission end of the diagram, as shown in fig. 1292. Next, with a radius equal to three times the length of the card (since the connecting-rod is six times the length of the crank), describe arcs from the points on the circumference of the circle to the diameter line, with centre on this diameter produced as shown, 90° – 90°, fig. 1292. From the divisions 10°, 20°, 30°...... on the diameter erect perpendiculars to the atmospheric line. The diagram is now divided up so that the pressure of steam per square inch of piston can be measured at every 10° of the revolution.

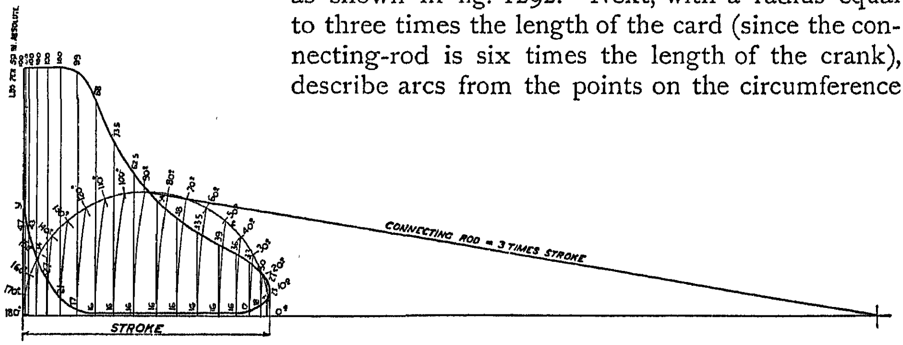


Fig. 1293

Taking, next, the diagram coming from the side of the piston *nearest* the crank-shaft, construct fig. 1293 in a similar manner, but with the arcs struck from centres on the opposite side, as shown.

The next step is to transfer the pressures on the piston to diagrams in which the base represents one complete revolution of the crank. Fig.

1294 is the reproduction of fig. 1292, and fig. 1295 of fig. 1293. The angles are measured from the dead-centre nearest to the cylinder. Now for the case of a single-acting simple engine such as the Willans, where the end of the cylinder away from the shaft is the working end, that nearest the shaft being exposed to the exhaust only, fig. 1294 represents the pressure per square inch of piston area exerted by the steam and transmitted to the cross-head pin by the piston-rod. Besides the pressures on the piston due to the steam there is another force acting on the moving parts of the engine when running, viz. the inertia of the moving masses, and this force will modify the pressures transmitted to the crank-pin to a greater or less extent according to the velocity of rotation, the length of crank, and the weight of the reciprocating parts of the engine. In high-speed engines it is often possible to keep down the weight of piston, piston-rod, cross-head, and connecting-rod to an amount equivalent to 2 lbs. per square inch of piston.

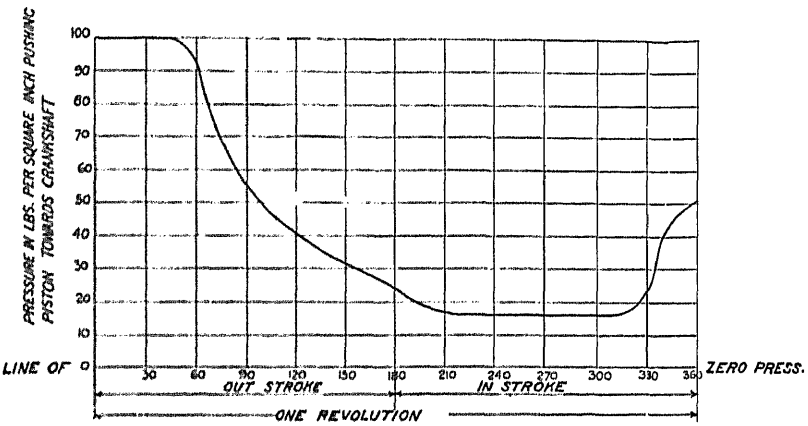


Fig. 1294

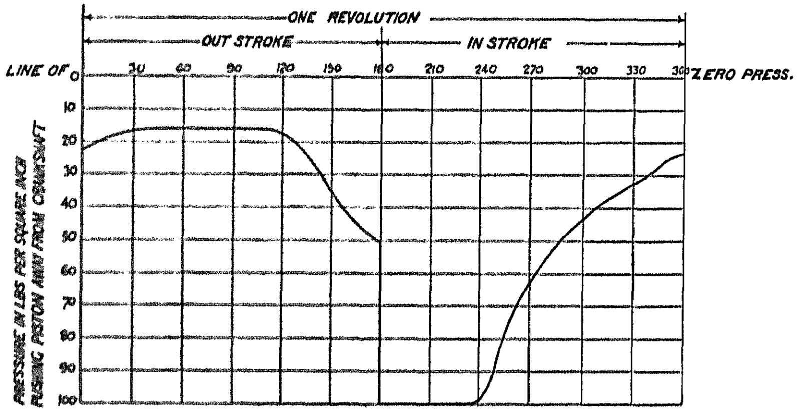


Fig. 1295

An approximate value of the inertia force, in the direction of the axis of the cylinder, may be obtained by calculating the centrifugal force of the weight of the reciprocating parts, assuming it to be concentrated at the crank-pin, adding one-sixth of this amount to the centrifugal force for the position when crank is passing the *near* dead-centre, and deducting one-sixth of the centrifugal force for the position when the crank passes the *far* dead-centre. This is for the case when the connecting-rod is equal to *six* times the crank-arm. Further, for the position of the crank when the piston is at mid-stroke, the inertia force will be equal to one-sixth the total centrifugal force, and acts in the direction *from* the cylinder towards the crank-shaft. An arc of a circle passing through these three points gives an approximate value of the inertia force (see fig. 1296).

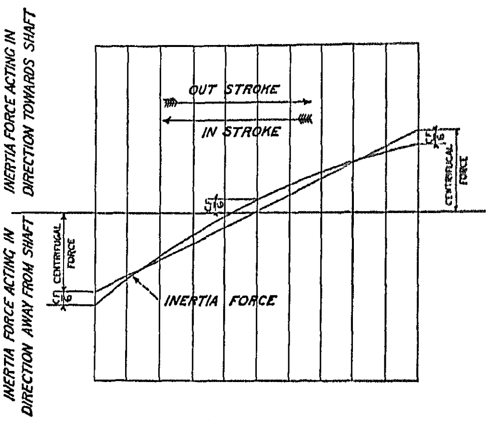

INERTIA FORCE ACTING IN DIRECTION TOWARDS SHAFT
INERTIA FORCE ACTING IN DIRECTION AWAY FROM SHAFT
OUT STROKE
IN STROKE
C.F./6
C.F./6
C.F./6
CENTRIFUGAL FORCE
CENTRIFUGAL FORCE
INERTIA FORCE

Fig. 1296

If this diagram is divided up in the same way as the indicator diagram was divided, into degrees of the revolution, values of the inertia forces are obtained for every 10° of the revolution. Table I gives the values so obtained

TABLE I

At 360° and 0° the inertia force = − 1.1665 × centrifugal force

,, 350° ,, 10° ,, ,, ,, = − 1.1420 × ,, ,,

,, 340° ,, 20° ,, ,, ,, = − 1.0676 × ,, ,,

,, 330° ,, 30° ,, ,, ,, = − 0.9508 × ,, ,,

,, 320° ,, 40° ,, ,, ,, = − 0.7968 × ,, ,,

,, 310° ,, 50° ,, ,, ,, = − 0.6149 × ,, ,,

,, 300° ,, 60° ,, ,, ,, = − 0.4166 × ,, ,,

,, 290° ,, 70° ,, ,, ,, = − 0.2134 × ,, ,,

,, 280° ,, 80° ,, ,, ,, = − 0.1130 × ,, ,,

,, 270° ,, 90° ,, ,, ,, = + 0.0791 × ,, ,,

,, 260° ,, 100° ,, ,, ,, = + 0.3322 × ,, ,,

,, 250° ,, 110° ,, ,, ,, = + 0.4708 × ,, ,,

,, 240° ,, 120° ,, ,, ,, = + 0.5834 × ,, ,,

,, 230° ,, 130° ,, ,, ,, = + 0.6710 × ,, ,,

,, 220° ,, 140° ,, ,, ,, = + 0.7361 × ,, ,,

,, 210° ,, 150° ,, ,, ,, = + 0.7817 × ,, ,,

,, 200° ,, 160° ,, ,, ,, = + 0.8115 × ,, ,,

,, 190° ,, 170° ,, ,, ,, = + 0.8283 × ,, ,,

,, 180° ,, ,, ,, = + 0.8335 × ,, ,,

In the above table the − sign means that the force acts in the direction from shaft towards cylinder.

Case I.—It is now possible to correct the diagrams in figs. 1294 and

1295 for inertia. Let it be assumed that the indicator diagram, fig. 1291, was obtained from a single-acting single-crank vertical high-speed engine running at 375 revolutions per minute, and having a 15-inch diameter cylinder and 10-inch stroke. In this case the working end of the cylinder is that farthest from the crank-shaft, hence fig. 1294 is the diagram of steam pressures, which has to be corrected for inertia. The centrifugal force of the parts, assumed concentrated at the crank-pin, will be obtained from the formula—

$$\text{Centrifugal force} = W\frac{\omega^2 r}{g} = \frac{W r N^2}{35200},$$

where W = the weight of parts per square inch of piston area (lbs.),
r = radius of crank (inches),
N = revolutions per minute.

For the case when $r = 5$ inches, $W = 2$ lbs. per square inch of piston, and $N = 375$—

$$\text{C.F.} = \frac{2 \times 5 \times 140625}{35200} = 39.95 \text{ lbs. per square inch of piston.}$$

$$\text{Say} = 40 \text{ lbs.}$$

The value of inertia force is given, therefore, in TABLE II:—

At	0° and 360°	... 40 × 1.1665 =	− 46.7
„	10° „ 350°	... 40 × 1.1420 =	− 45.7
„	20° „ 340°	... 40 × 1.1067 =	− 42.7
„	30° „ 330°	... 40 × 0.9508 =	− 38.0
„	40° „ 320°	... 40 × 0.7968 =	− 31.9
„	50° „ 310°	... 40 × 0.6149 =	− 24.6
„	60° „ 300°	... 40 × 0.4166 =	− 16.7
„	70° „ 290°	... 40 × 0.2134 =	− 8.5
„	80° „ 280°	... 40 × 0.0154 =	− 0.62
„	90° „ 270°	... 40 × 0.1694 =	+ 6.77
„	100° „ 260°	... 40 × 0.3322 =	+ 13.3
„	110° „ 250°	... 40 × 0.4708 =	+ 18.8
„	120° „ 240°	... 40 × 0.5834 =	+ 23.3
„	130° „ 230°	... 40 × 0.6710 =	+ 26.8
„	140° „ 220°	... 40 × 0.7361 =	+ 29.4
„	150° „ 210°	... 40 × 0.7817 =	+ 31.3
„	160° „ 200°	... 40 × 0.8115 =	+ 32.5
„	170° „ 190°	... 40 × 0.8283 =	+ 33.1
„	180°	... 40 × 0.8335 =	+ 33.3

The sign
+ denotes a force acting in the direction from cylinder to shaft.
− denotes a force acting in the direction from shaft towards the cylinder.

By plotting these values on fig. 1293 and performing the addition, fig. 1297 is obtained, on which the heavy line gives the resultant vertical pressure transmitted along the axis of the piston-rod. This pressure resolves itself in two directions, one normal to the cross-head guide-bars, the other along the axis of the connecting-rod on to the crank-pin. The above assumes that the whole weight of the connecting-rod is carried by the cross-head, and is subject to the acceleration and retardation due to the motion of the cross-head, whereas in reality only the upper part of the connecting-rod has the motion of the cross-head, the big end having the motion of the crank-pin.

If it is desired to find the pressure between the big end brasses and crank-pin, it is more correct to consider the inertia of piston and rod and cross-head, plus one-third the connecting-rod as partaking of the reciprocating motion of the cross-head, and to add the centrifugal force of two-thirds the connecting-rod resolved in the direction of the axis of the cylinder. The method of taking the whole connecting-rod weight as partaking of the motion of the cross-head is, however, shorter, and does not introduce any considerable error. The centrifugal force of the big end of connecting-rod, however, has no power to influence the turning moment of an engine, since it acts along the centre of the crank-arm.

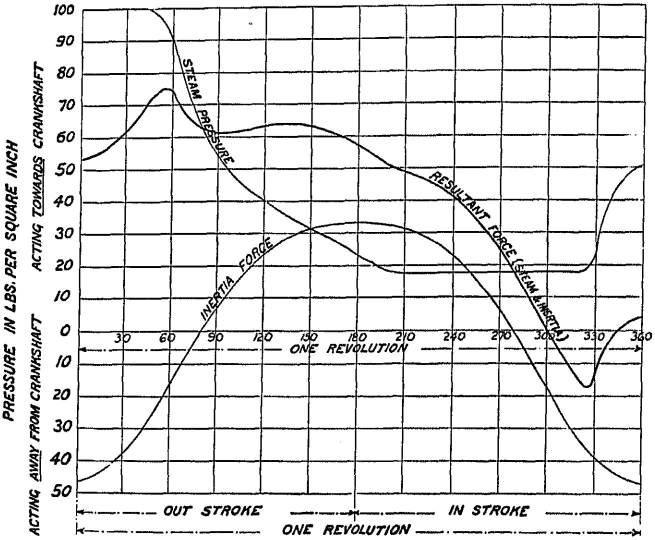


Fig. 1297

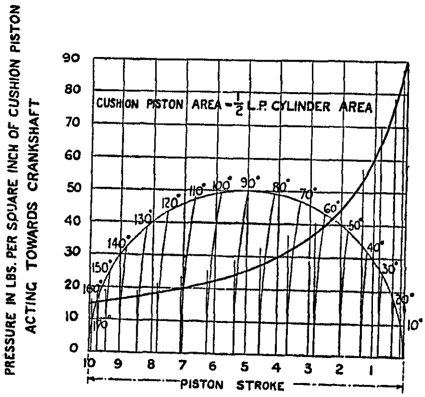


Fig. 1298

For single-acting engines, provision has to be made to prevent the inertia forces exceeding the steam pressure on the piston, in order that the connecting-rod may always be in *thrust* and the big end and crank-pin always in contact, so that the engine may run noiselessly. For this reason the inertia of the whole rod must first be taken into account in order that the proper amount of cushioning on the up-stroke may be arranged for. It will be seen in fig. 1297 that the resultant of inertia and steam pressure is an upward force, acting from 300° to about 345° in the revolution, and consequently some cushioning pressure must be added in order that "constant thrust" on the rods may be obtained. In Messrs. Davey, Paxman, & Co.'s "Peache" engine a steam-cushioning cylinder is provided, and in the Willans engine an air-cushioning cylinder. Fig. 1298 represents the pressure in such cushioning cylinders, and fig. 1299 is the reproduction of fig. 1297, but with the cushion pressure added.

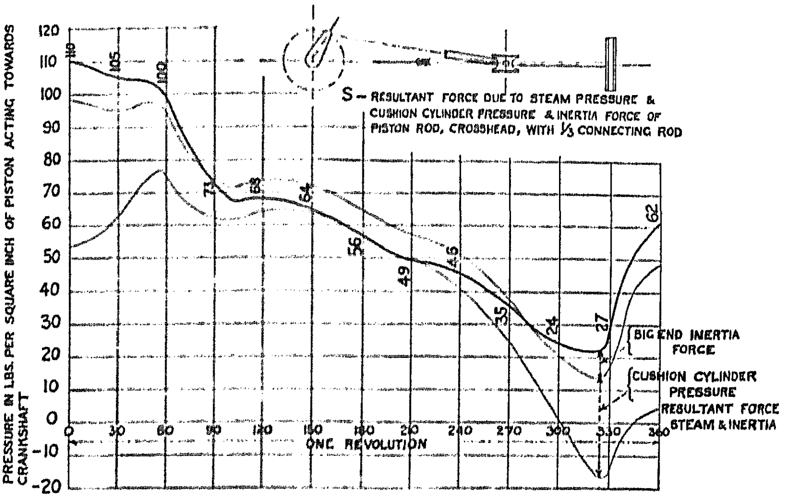


Fig. 1299

As has been stated, the big end of the connecting-rod and a portion of the rod partakes of the motion of the crank-pin, and cannot assist or retard the turning of the crank. If the total weight of piston, rod, cross-head, and connecting-rod is equal to 2 lbs. per square inch of the piston, the weight of piston, rod, cross-head, and one-third connecting-rod may amount to about 1½ lb. per square inch of piston. For the purpose of finding the resultant force S that affects *turning*, therefore, one-fourth of the inertia force in fig. 1297 must be deducted from the resultant force of the steam, the cushion, and the whole inertia in fig. 1299. Thus the final resultant for *turning* is found, as shown on fig. 1299, by the heavy line.

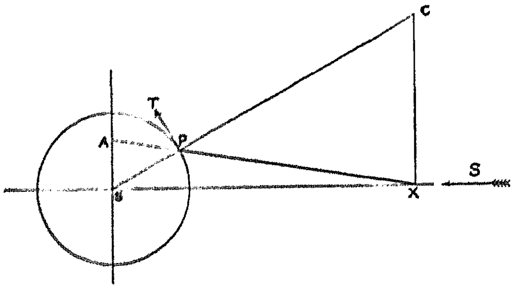


Fig. 1300

Having found the force acting along the axis of the piston-rod, it is necessary to find what is the value of the force which acts on the crank-pin *tangentially* to the crank-pin path. To do this the well-known graphic method is perhaps that most generally used by practical engineers. In fig. 1300, let PX represent the connecting-rod and SP the crank of an engine. P the crank-pin rotates about the shaft S. The cross-head X slides along the axis XS, and during the moment it is at the point X may be considered as rotating about some point C in a line through X T, perpendicular to the axis SX. In fact, during the instant when the crank-pin is at P and cross-head at X, the connecting-rod is said to be rotating about the instantaneous centre of rotation C, which is at the intersection of the perpendicular through X and the line SP produced, for at that moment the path of P is tangential to a circle about C as centre, as is also the path of X. From this it follows that—

$$\frac{\text{Velocity of X}}{\text{Velocity of P}} = \frac{\text{CX}}{\text{CP}} = \frac{\text{SA}}{\text{SP}} \text{ (by similar triangles).}$$

Since PX the connecting-rod is a rigid body acted on at X by a force S in the direction XS, it follows that the pressures transmitted at P and X are inversely as the velocities of P and X, hence—

$$\text{T} = \text{S}\,\frac{\text{velocity of X}}{\text{velocity of P}} = \text{S} \cdot \frac{\text{SA}}{\text{SP}}.$$

It is possible now, by constructing fig. 1300 to scale, and measuring the length SA and SP, to compile a table giving the portion of the force S from fig. 1299, which acts tangentially to the crank-pin circle, and is the turning effort on the crank-pin.

TABLE III

Angle from Commencement of Stroke from Dead-Centre Nearest the Cylinder.	$\frac{\text{SA}}{\text{SP}}$ =
0°	0.000
10° and 350°	0.202
20° „ 340°	0.396
30° „ 330°	0.572
40° „ 320°	0.725
50° „ 310°	0.849
60° „ 300°	0.939
70° „ 290°	0.994
80° „ 280°	1.014
90° „ 270°	1.000
100° „ 260°	0.956
110° „ 250°	0.885
120° „ 240°	0.793
130° „ 230°	0.683
140° „ 220°	0.560
150° „ 210°	0.427
160° „ 200°	0.288
170° „ 190°	0.145
180°	0.000

From Table III and fig. 1299 the amount of the crank effort for every 10° of the revolution can be obtained, and these values, plotted on a diagram having the base-line divided up to represent the degrees of a revolution, give the crank-effort diagram, fig. 1301, which shows the amount of the tangential pressure on the crank-pin at every position of the crank-pin in a revolution.

As the case under consideration is that of a single-crank single-acting engine, it is readily seen that during the outward stroke this effort is a *driving* effort in the direction of rotation, and during the inward stroke is a *retarding* effort. Driving efforts are plotted above the line of zero pressure, and retarding efforts below that line. Now the resistance to turning offered by a dynamo is constant throughout a revolution, but the turning effort of the engine is seen by fig. 1301 to be irregular. During the outward or driving stroke the steam has to exert sufficient energy not only to balance the dynamo resistance for one complete revolution, but also to balance the resistance offered by the pressure of the exhaust steam and compression in the cylinder on the inward stroke. The excess of energy exerted by the steam on the driving stroke is stored in the fly-wheel and armature of the dynamo, and restored by them on the inward stroke.

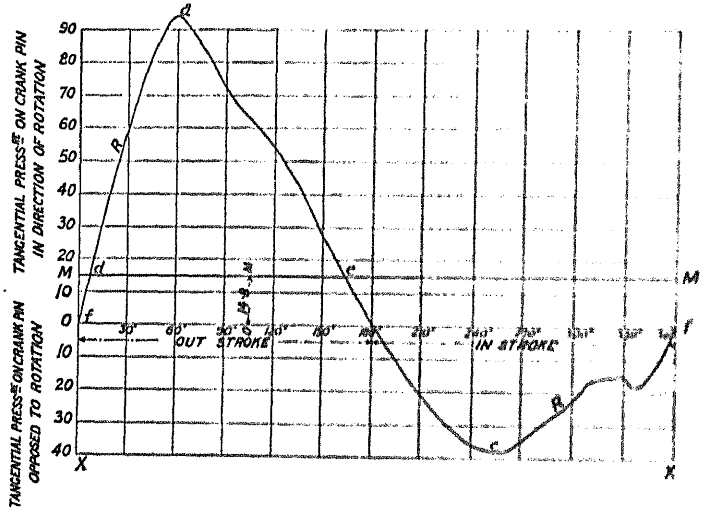


Fig. 1301.—Crank-Effort Diagram. Single-Acting High-Speed Engine—Single Crank.

Coefficient of fluctuation $= \frac{8.64}{6.66} = 1.297$

The *mean* crank effort can be found by drawing a line X X in fig. 1301 parallel to O O, and measuring the area X O *a c* O X by planimeter or otherwise. Dividing this area by the length of the base O O gives a height X M such that X M × O O = area X O *a c* O X. O M is the *mean crank effort*. The area X M × O O gives, to scale, the number of foot-pounds of energy exerted by the engine per revolution E_0. If the engine is running steadily the mean crank effort O M is equal to the resistance to turning offered by the dynamo and friction. The excess energy of the steam represented by the area *dae* is stored in the fly-wheel and armature, and results in an increase of speed which attains a maximum at *e*. After that the fly-wheel and armature restore this energy, thereby losing speed, which reaches its lowest limit at *d*.

The ratio—

$$\frac{\text{Excess energy}}{\text{Mean energy per revolution}} = \frac{\text{area } dae}{\text{area MMOO}} = \frac{e}{E_0} = \kappa$$

is called the "coefficient of fluctuation of energy", and in this case

$$\kappa = 1.297.$$

Case II.—The next case taken is that of a single-acting two-crank engine, with cranks at 180°, with a single cylinder operating each crank, the indicator diagram from each cylinder being that given in fig. 1292. The coefficient of fluctuation of energy will be obtained by combining two crank-effort diagrams from fig. 1301, the second diagram starting at 180° of the revolution, since the cranks are at 180° (see fig. 1302). The *mean* crank effort is found as before, RRR representing the tangential pressure at the radius of the crank per square inch of *one* piston, but due to both pistons. Here it is seen that the excess energy (area *dae*) is very much less than in the case of the single-crank engine, and not only that, but the mean energy per revolution of the engine is twice that of the single-crank engine. On measuring these areas as before, it will be found that the coefficient of fluctuation $\dfrac{\text{area } dae}{\text{area MMOO}}$

$$= \kappa = 0.134.$$

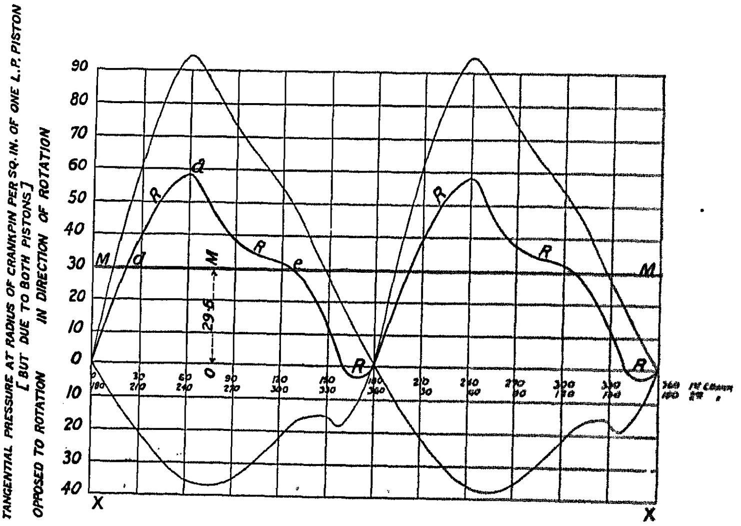


Fig. 1302.—Crank-Effort Diagram. Single-Acting High-Speed Engine—Two Cranks. Coefficient of fluctuation = 0.134

Case III.—The last case it is intended to take for single-acting engines is that of a three-crank engine with a single cylinder operating each crank, the cranks being at 120° to each other. Fig. 1303 gives the combination of *three* crank-effort diagrams as in fig. 1301, the second starting at 120°, and the third at 240° from the first. Here $\kappa = 0.064$.

Coming now to double-acting high-speed engines, *Case IV* is that of a single-crank double-acting single-cylinder engine. Assuming fig. 1291 represents the indicator diagram obtained from each side of the piston, the driving pressure on each square inch of piston area will be found by deducting the exhaust and compression line of the card from the one side, from the admission and expansion line of the card on the other side of the piston. By combining figs. 1294 and 1295, fig. 1304 is obtained, giving the resultant driving pressure of the steam per square inch of piston. Assuming that the weight of the piston, piston-rod, cross-head, and connecting-rod in this engine is again equivalent to 2 lbs. per square inch of piston area, as in the single-acting engine, fig. 1304 can be corrected for inertia, the speed and stroke being again taken as 375 revolutions per minute, and 10 inches respectively. Table II can be used again, remembering, as it is turning effort that is under consideration, to take only three-fourths of the values given in Table II, since the "big end" of the connecting-rod revolves with the crank-pin, and its inertia cannot affect the turning effort. By multiplying the pressures given on fig. 1304 at every 10° of the revolution, by the figures given in Table III as before, fig. 1305 is obtained, which gives the crank-effort diagram of a single-crank double-acting high-speed engine. Here $\kappa = 0.143$.

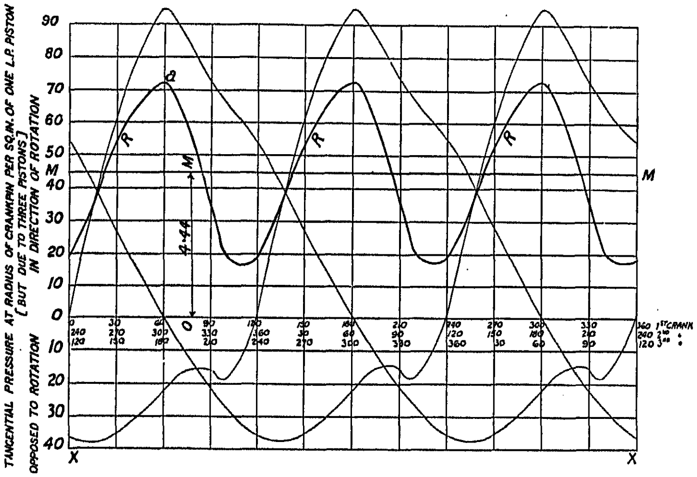


Fig. 1303.—Crank-Effort Diagram. Single-Acting High-Speed Engine—Three Cranks.
Coefficient of fluctuation = 0.064

Case V is that of a two-crank double-acting engine with a single cylinder operating each crank. Cranks at 180°. As in this case the inequality of the inertia forces due to the angularity of the connecting-rod in the two semi-revolutions is wiped out by means of the two cranks at 180°, the coefficient of fluctuation is slightly less than in Case IV. The value of κ is 0.132.

Case VI is that of a three-crank double-acting engine with a single cylinder operating each crank. Cranks at 120°. To construct the crank-effort diagram for this case, plot fig. 1305 for the first crank; next, beginning at 120° on the base, plot it again for the second crank; and beginning at 240°, plot it a third time for the third crank. Adding the three curves, the resultant turning effort R R R is found. The mean turning effort is found as before, and the excess energy *d a e* measured. Here $\kappa = 0.011$. See fig. 1306.

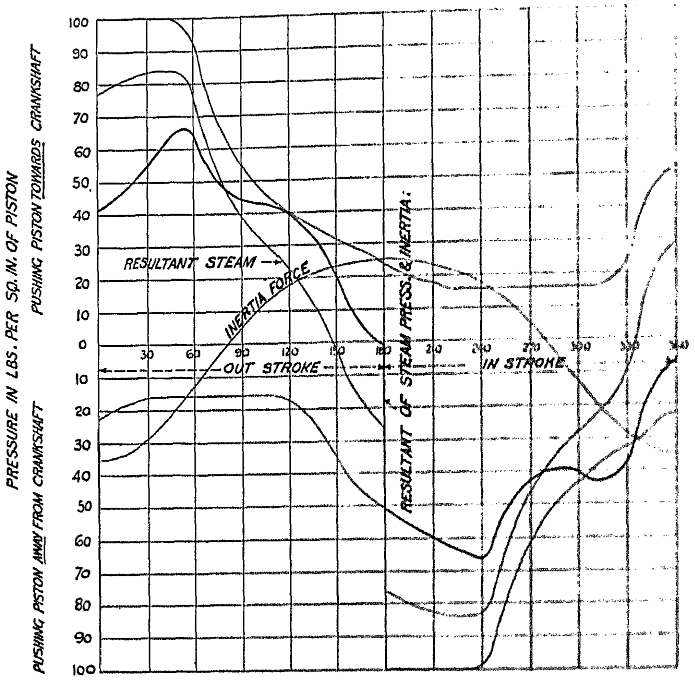


Fig. 1304

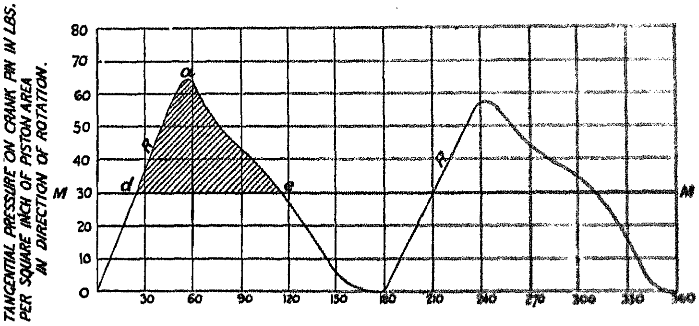


Fig. 1305.—Crank-Effort Diagram. Double-Acting High-Speed Engine—Single Crank
Coefficient of fluctuation = 0.143

In slow-speed engines the reciprocating parts are as a rule much heavier per square inch of piston area than in high-speed engines. As cylinders become bigger, for the same initial steam pressure it is no longer possible to keep the same margin of strength without adding to the weight. In slow-speed engines, both the stroke and diameter of pistons are necessarily larger for a given power than in high-speed engines. The stroke being longer, the piston-rods and connecting-rods, even for the same total load, have to be made bigger in diameter in order that they may have sufficient stability as columns. As the load on the piston-rod is also greater in slow-speed engines than in high-speed engines of the same power, it follows that the rods will be bigger in diameter, both for stability as columns and also to carry the greater load.

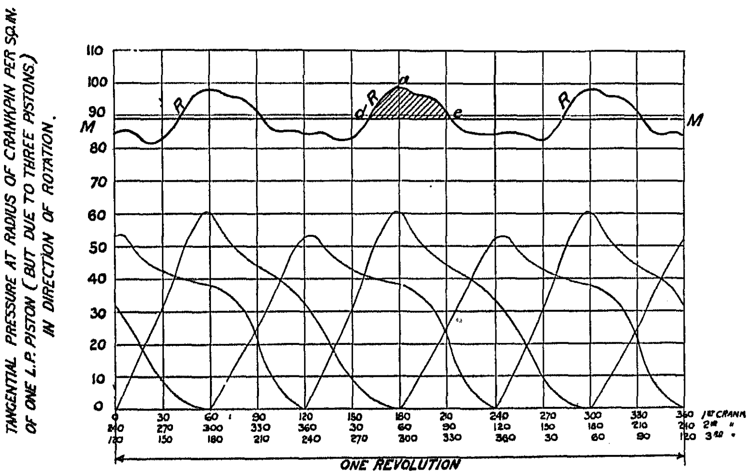


Fig. 1306.—Crank-Effort Diagram. Double-Acting High-Speed Engine—Three Cranks. Coefficient of fluctuation = 0.011

It must not be understood that in all high-speed engines the weight of parts can be kept down to the 2 lbs. per square inch of piston area that has been assumed for the purpose of the previous calculations; for small engines it is sometimes as low as 1½ lb. per square inch of piston. The weight increases as the size of the piston increases, and may be as much as 3 lbs. per square inch in engines running at about 250 revolutions per minute. In some engines it is even more than this.

Coming to the case of what are known as slow-speed engines, running at from 60 to 120 revolutions per minute, the weight of the reciprocating parts may be kept down (with proper regard to strength) to from 4 to 6 lbs. per square inch of piston area.

In preparing the crank-effort diagrams for slow-speed simple-expansion engines, having the cut-off in the cylinder arranged so that the indicator diagram fig. 1291 is obtained from each end of the cylinder, the only alteration to be made to the previous diagrams is that necessary to show the effect of the altered inertia forces. After having done that the process is the same as has been described.

Case VII.—Slow-speed double-acting engine. Single crank. Connecting-rod = 6 cranks. 15 inches diameter cylinder. 24 inches stroke. 140 revolutions per minute. Weight of reciprocating parts taken as 4 lbs.

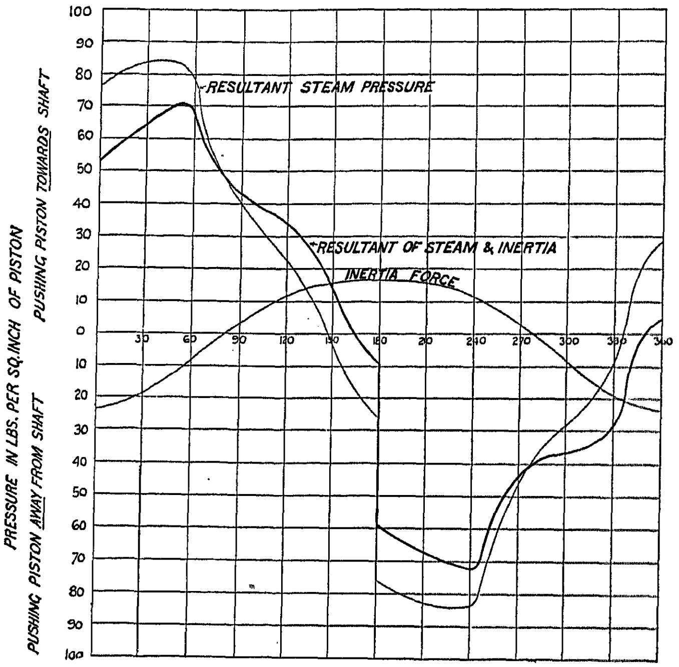


Fig. 1307

per square inch of piston area. Taking, as before, only one-third the weight of the connecting-rod in addition to that of the piston, rod, and cross-head, a table can be compiled from Table II to get the value of the inertia forces.

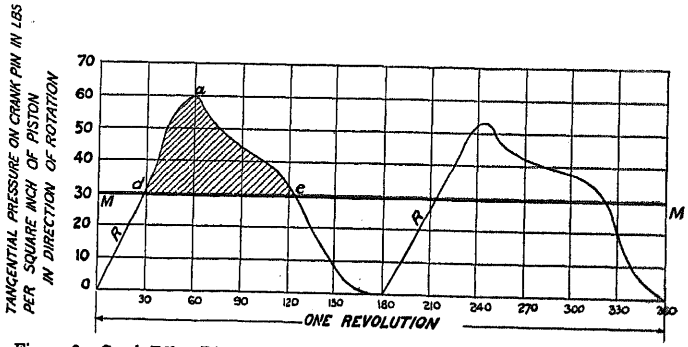


Fig. 1308.—Crank-Effort Diagram. Double-Acting Slow-Speed Engine—Single Crank. Coefficient of fluctuation = 0.1485

Total weight of reciprocating parts = 4 lbs. per square inch. Weight of parts whose reciprocation affects turning = 3 lbs. per square inch of piston area. Centrifugal force $= \dfrac{3 \times 12 \times 140^2}{35200} = 20$ lbs. (about).

Evidently the inertia forces for this engine are half those given in Table II.

Fig. 1307 is constructed as before, showing the resultant pressures due to steam and inertia. By applying the figures in Table III, fig. 1308 is drawn, showing the crank effort of this engine. Measurement of the areas *d a e* and O M M O gives the coefficient of fluctuation of energy—

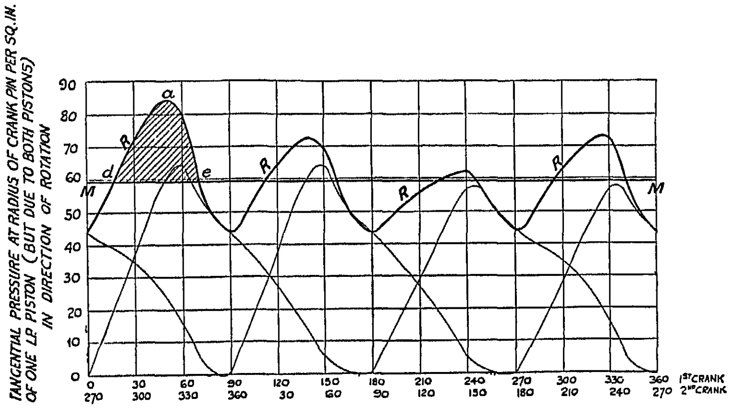


Fig. 1309.—Crank-Effort Diagram. Double-Acting Slow-Speed Engine—Two Cranks at 90°
Coefficient of fluctuation = 0.0394

$$\kappa = \frac{d\,a\,e}{\mathrm{OMMO}} = \frac{e}{\mathrm{E}_0} = 0.1485$$

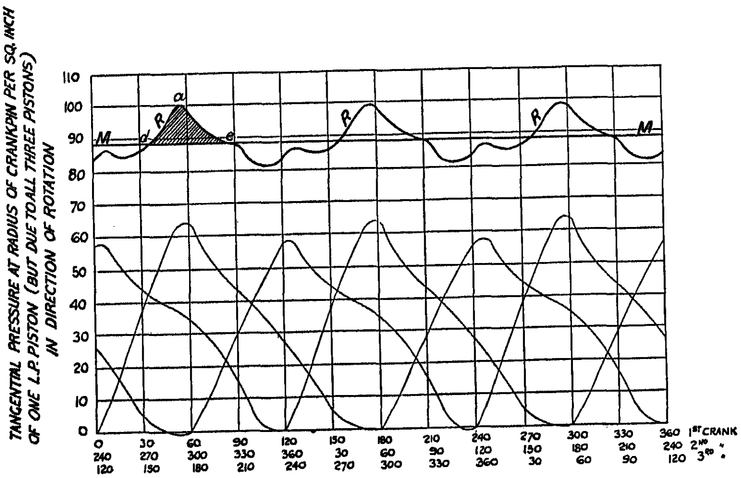


Fig. 1310.—Crank-Effort Diagram. Double-Acting Slow-Speed Engine—Three Cranks at 120°.
Coefficient of fluctuation = 0.0095

Case VIII.—Slow-speed double-acting engine. Two cranks at 90°. Single cylinder operating each crank, or cross-compound engine, if powers are *absolutely* evenly divided between H.P. and L.P. cylinders. The crank-effort diagram, fig. 1309, is obtained by combining two curves from fig.

1308, making the second curve start when first crank has reached 90° in the revolution. Coefficient of fluctuation of energy: $\kappa = 0.0394$.

Case IX.—Slow-speed double-acting engine. Three cranks at 120°. Single cylinder operating each crank. Simple expansion, or triple expansion H.P., I.P., L.P. on the separate cranks if power is *absolutely* evenly divided between the three cylinders. Fig. 1310 is obtained by the combination of three curves from fig. 1308. Coefficient of fluctuation of energy: $\kappa = 0.0095$.

Collecting the results for all engines considered, the coefficients of fluctuation of energy may compared as below:—

Type of Engine.	Coefficient of Fluctuation of Energy expressed as a fraction of the engine energy *per revolution.*
High-Speed Single-Acting—	
Case I.—Single crank	1.297
Case II.—Two cranks at 180° ...	0.134
Case III.—Three cranks at 120° ...	0.064
High-Speed Double-Acting—	
Case IV.—Single crank	0.143
Case V.—Two cranks at 180° ...	0.132
Case VI.—Three cranks at 120° ...	0.011
Slow-Speed Double-Acting—	
Case VII.—Single crank	0.148
Case VIII.—Two cranks at 90° ...	0.039
Case IX.—Three cranks at 120° ...	0.0095

It must be borne in mind that the above figures are true only for simple-expansion engines having a single cylinder operating each crank. Compound engines and triple-expansion engines are chiefly in use, and for such cases, where each crank is driven by a complete compound or triple-expansion engine with the cylinders placed tandem fashion on each crank, the coefficients of fluctuation of energy will be very much the same as for the simple-expansion engines worked out. For compound and triple-expansion engines in which the cylinders are placed side by side, the above figures will only hold where the H.P. and L.P. cylinders are so proportioned, and the cut-off of the steam so adjusted in each cylinder, that the total work of the engine is equally divided amongst the several cylinders, and the weights of the reciprocating parts equal on each crank. This is rather a difficult state of affairs to obtain, and even when obtained is likely to be upset at any time by a leaky valve allowing steam to pass direct from the steam-chest to the intermediate receiver, and thus cause the L.P. cylinder to do more work than is intended. For engines governed by throttle-valve, equality of work between the several cylinders can only be obtained at one particular load, at all other loads the work is bound to be unevenly divided between the cylinders. Fig. 1311 shows the crank-effort diagram of a double-acting side-by-side engine in which the powers exerted are as 26.7 per cent on 1st crank, 33.3 per cent on 2nd crank, 40.0 per cent on 3rd crank.

The coefficient of fluctuation is in this case $\kappa = 0.033$.

Not only is the coefficient of fluctuation three times that of a triple-expansion engine with complete engine on each crank, but it will be noticed also that there are only *two* virtual impulses per revolution. The effect of this will be touched upon later. In the case of a complete engine driving each crank, a leaky valve will also upset the evenness of turning, but not to nearly the same extent as in the case of a side-by-side engine; the leakage will only be a part of the steam passing through one engine, and not the same proportion of the whole steam passing through the whole engine, as in the case of a side-by-side engine.

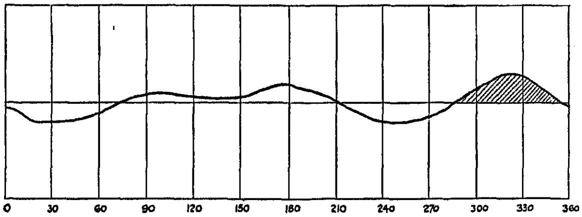


Fig. 1311, Case X.—Crank-Effort Diagram. Double-Acting Three-Crank Side-by-Side Triple-Expansion High-Speed Engine—Cranks at 120°. Coefficient of fluctuation = 0.033

Fly-Wheel Energy.—Having seen now how the turning effort may vary between different types of engines, and how the excess energy over the mean energy results in an increase of speed, it remains to be considered how this resulting variation of speed during the revolution may be kept within bounds. The well-known formula for the kinetic energy of a moving mass is—

$$\text{Energy } E = \tfrac{1}{2} M v^2,$$

where M is the mass of the moving body and v its velocity. Inspection of this formula shows that any addition of energy to that already possessed by the moving parts of the engine must result in increase of speed, since the mass of the moving parts does not vary. The shaft, big end of connecting-rod, armature (or rotating magnet wheel), together with the engine fly-wheel, if any, may be taken as together equivalent to a certain virtual fly-wheel possessing energy $E = \frac{1}{2} M v^2$, M being the mass of the rim of this virtual fly-wheel and v the velocity of this rim.

If V_H and V_L are taken to denote the highest and lowest velocities of this rim, due to the storage and restorage of energy in and by it, V being the mean velocity, the fluctuation of energy of the wheel is given by the equation—

$$\Delta E = \kappa E_0 = \frac{M}{2} (V_H^2 - V_L^2).$$

Here $\kappa E_0 = e$, the excess energy over the mean energy, E_0 being the energy exerted by the engine per revolution. Inspection of this formula shows that for a certain value of e or κE_0, if $V_H - V_L$ is to be kept low, M must be considerable The bigger M the smaller the variation of speed.

Further—

$$\begin{aligned} V_H^2 - V_L^2 &= (V_H + V_L)(V_H - V_L) \\ &= 2\,V\,(V_H - V_L). \end{aligned}$$

Hence—

$$e = \kappa E_0 = M\,V\,(V_H - V_L).$$

Permissible Fluctuation of Speed.—The fluctuation of speed $\frac{V_H - V_L}{V} = q$ which can be allowed depends upon the nature of the work to be done and the type of generator in use, as well as the type of engine. For continuous-current generators it is not of such great importance to have very even turning, but for alternator work it is absolutely necessary to keep the fluctuation of speed low. The reason why such regularity of turning is required is an electrical one, and may be briefly stated as follows:—

For electrical reasons a certain phase difference between two alternators running in parallel must not be exceeded. Taking the case of two 2-pole alternators running in parallel, and assuming that the allowable phase displacement between them is 2°, it follows that no pole on the magnet wheel of either alternator may be out of the position it would occupy if turning absolutely evenly by more than 1°.

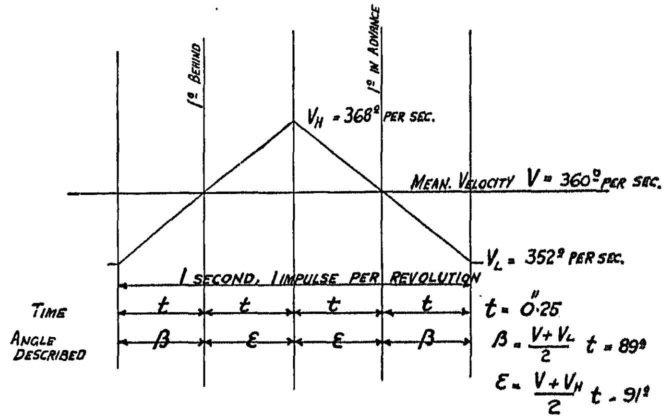

1° BEHIND
1° IN ADVANCE
V_H = 368° PER SEC.
MEAN. VELOCITY V = 360° PER SEC.
V_L = 352° PER SEC.
1 SECOND, 1 IMPULSE PER REVOLUTION
TIME
t
ANGLE DESCRIBED
β
ε
$t = 0''.25$
$\beta = \frac{V+V_L}{2}\,t = 89°$
$\varepsilon = \frac{V+V_H}{2}\,t = 91°$

Fig. 131a

A 2-pole alternator gives 1 alternation each revolution.
A 4- " 2 " "
A 2p- " p " "

If $a°$ is the angular displacement permitted in a 2-pole alternator, $\frac{a}{p}$ will be the angular displacement allowable for a multipolar alternator, p being the number of pairs of poles.

Hence it is seen that the greater the number of poles, the smaller must the angular displacement of the magnet wheel be from the position it would occupy if rotating absolutely evenly during the revolution.

During one revolution each pair of poles of an alternator describes 360°, therefore the number of actual degrees described by a magnet pole during one complete alternation is $360 \div p$. The 1° allowed angular displacement will occur at the beginning and end of each impulse given by the engine, provided the duration and magnitude of the impulses are all equal. If the engine gives one impulse each revolution, the magnet wheel will be 1° behind its proper position once in each revolution, and also 1° in advance of its proper position once in each revolution, and these two points will occur at 180° apart.

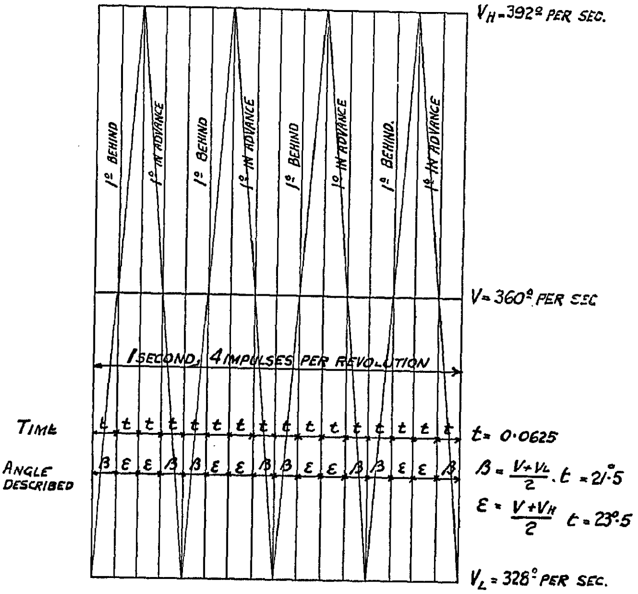


Fig. 1313

Assume for the moment that the velocity of the wheel rises and falls as shown by the straight lines in fig. 1312. In order that the wheel may not travel more than 91°, while an absolutely regularly turning wheel would describe 90°, the maximum velocity must not exceed 368° per second in an engine describing 360° per second as a mean velocity, nor must it sink to less than 352° per second, consequently—

$$q = \frac{V_H - V_L}{V} = \frac{16}{360}.$$

If the engine gives *four* impulses per revolution, the maximum velocity must not exceed 392° per second, or the minimum be less than 328° per second; hence in this case—

$$q = \frac{V_H - V_L}{V} = \frac{64}{360}. \quad \text{(See fig. 1313.)}$$

From this it is evident that the greater the number of impulses the greater may be the fluctuation of speed for the same angular displacement. The actual shape of the velocity curves will, however, be determined by the shape of the impulse curves on the crank-effort diagrams, and consequently the maximum and minimum velocities can only be determined for each individual case from the crank-effort diagram. It is certain that the speed fluctuation need never be less than that shown in fig. 1314, where the velocity is supposed to jump up instantaneously to the mean value of the *excess velocity*, and to sink also instantaneously to the mean value of the deficient velocity. The *limit value of q*, therefore, will be—

$$= \frac{8}{360} \text{ for the case shown in fig. 1312, and}$$

$$= \frac{32}{360} \text{ for the case shown in fig. 1313.}$$

Further, the impulses of an actual engine are seldom, if ever, of equal duration, but if they were equal the above equations could be made to include the number of impulses per revolution by writing them—

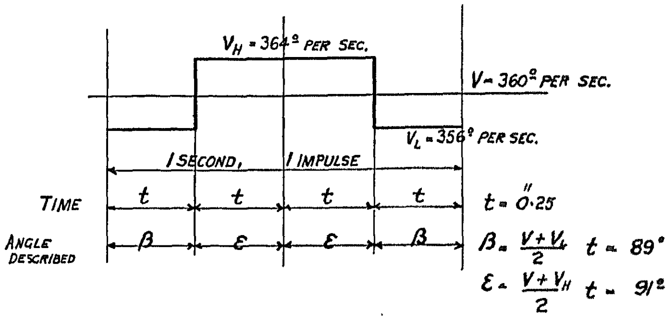


Fig. 1314

$$q = \frac{4 \times 2n}{360}.$$

Also, to include the effect of the number of poles on the magnet wheels the equation may be written—

$$q = \frac{4 \times 2n \times \alpha}{360 \times p}.$$

Inasmuch as the duration of the impulses varies considerably, a *limit value* for q may be arrived at which will certainly cover any conceivable case, if instead of taking twice the number of impulses in the equation, the actual duration θ of the biggest impulse is included. For $2n$ write $\frac{360}{\theta}$. The equation then becomes—

$$\text{Limit value of } q = \frac{4 \cdot \alpha}{\theta \cdot p},$$

where α is permissible angular displacement,
θ duration in degrees of the biggest impulse,
p the number of *pairs* of poles.

Let it be assumed now that any of the engines for which the coefficient of fluctuation of energy κ has been found has to be provided with sufficient momentum in the revolving parts to keep the angular displacement (from absolutely even turning) down to 1°, ÷ the number of pairs of poles.

Calculation of Fly-Wheel Energy for—

$\phi = 2°$ phase displacement between two alternators,
i.e. $\alpha = 1° \div p$ angular displacement.

It has been shown—

$$\kappa E_0 = \frac{M}{2}(V_H - V_L)(V_H + V_L),$$

$$\text{and } q = \frac{V_H - V_L}{V}.$$

Therefore—

$$\kappa E_0 = \frac{M}{2} q V(V_H + V_2) = 2\frac{M V^2}{2} q.$$

$$\kappa E_0 = 2 E q.$$

This last equation gives all that is needful for determining the amount of energy required in the equivalent fly-wheel, which is in reality supplied by magnet wheel and engine fly-wheel, if any.

By measuring the indicator diagram, fig. 1291, in the usual way it is found that the *mean effective pressure* of the steam on the piston is 46.5 lbs. per square inch of piston area. This, by the assumption made that the diagram fig. 1291 is common to all the cylinders of all the engines under consideration, is the mean pressure that must be taken in all calculations for Cases I to IX to obtain the I.H.P. of the various engines.

It is not necessary to give the calculations in detail for more than one example—Case III, Table IV has been chosen for this purpose. The results of all cases taken are given in Table IV.

Case III.

Single-acting high-speed engine.
Three cranks at 120°.
300 kilowatts output. Periodicity, 50.
481 I.H.P.
$\theta = 65°$, 3 virtual impulses per revolution. (See fig. 1303.)
300 revolutions per minute.
Cylinder diameter, 21⅛ inches; stroke, 13 inches.

$$\text{I.H.P.} = \frac{46.5 \times 39 \times 350 \times 300}{12 \times 33000} = 481.$$

$$E_0 = 52800 \text{ ft.-lbs.}$$

$$\kappa = 0.064 \text{ (fig. 020).}$$

$$\kappa E_0 = 3379 \text{ ft. lbs.}$$

$$p = \frac{50 \times 60}{300} = 10.$$

$$q = \frac{4a}{\theta p} = \frac{4}{65 \times 10} = \frac{1}{162.5}.$$

$$E = \frac{\kappa E_0}{2q} = 274543 \text{ ft.-lbs.}$$

$$\frac{E}{E_0} = 5.2.$$

W at 80 feet per second = 2910 lbs.
1.3 ton.

In double-acting high-speed engines the speed at which the engines can be run is limited by the inertia forces. These must not be so great as to cause impulses on the cranks in a direction contrary to that of rotation. The speeds chosen in the following figures are such as would be realized in engines running with forced lubrication (the invention of which has made high speed possible for double-acting engines).

Cases I to IX have been calculated for single expansion of the steam in one cylinder on each crank. The coefficient of fluctuation of energy, and consequently the energy required in the fly-wheel, will vary according to the way the piston pressures (including inertia forces) are applied. It has been thought best, for the sake of comparison, to give the turning efforts for the several arrangements of cranks in general use, assuming the powers exactly equal on each crank and the cut-off and pressures on the pistons the same for each arrangement, and afterwards to recalculate the fly-wheel energy required from coefficients obtained from actual indicator diagrams.

Table IV shows the results obtained, in tabulated form, for comparison.

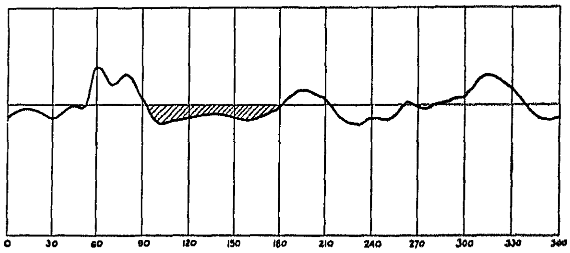


Fig. 1315, Case XI.—Crank-Effort Diagram. Single-Acting Compound Three-Crank Tandem High-Speed Engine, "Peache" Type—Cranks at 120°. Coefficient of fluctuation = 0.022

Case X is that of an actual example of a side-by-side triple-expansion engine. The cylinders are arranged H.P., I.P., L.P., one on each crank. In this case the powers are unevenly divided on the cranks, being as 80 : 100 : 120. Fig. 1311 gives the crank-effort diagram. It will be noticed that the inequality of powers on the cranks has the effect of giving only two virtual impulses on the generator. As the engine is a three-cylinder double-acting engine, there are, of course, six actual impulses, but of unequal strength.

Case XIV is of a slow-speed side-by-side triple-expansion engine. Here again the powers on the cranks are unequal, but not to such an extent as in Case X. The consequence is that there are four virtual impulses on the generator. Fig. 1318 shows the crank-effort diagram.

From these two cases it is seen that it by no means follows that a three-crank double-acting engine will give six impulses on the generator.

Case XI is an example of a special type of single-acting high-speed engine. Fig. 1315 is the crank-effort diagram. The engine has a complete compound engine on each crank, the powers in this case being as 102 : 100 : 98. As the engine has a complete set of compound cylinders above each crank, it is easy to keep the powers on the cranks equal. It will be noticed that the coefficient of fluctuation of energy is smaller than that given in Case III for a simple single-acting engine. The reason of this is that, owing to the peculiar arrangement of cylinders (the L.P. piston taking steam on its under side), the retarding influence of the back pressures on the upward stroke is much less than in the case of a single-acting engine, in which the pressure acts on the upper side of

TABLE IV

TYPE OF ENGINE.	Single-Acting High-Speed Simple Engines.			Double-Acting High-Speed Simple Engines.			Double-Acting Slow-Speed Simple Engines.			Double-Acting High-Speed Side-by-Side Triple Expansion.	Single-Acting High-Speed Tandem Compound Engine on each Crank.	Double-Acting Slow-Speed Cross Compound Engines.		Double-Acting Slow-Speed Side-by-Side Triple Expansion.
	One Crank.	Two Cranks at 180°.	Three Cranks at 120°.	One Crank.	Two Cranks at 180°.	Three Cranks at 120°.	One Crank.	Two Cranks at 90°.	Three Cranks at 120°.	Three Cranks at 120°.	Three Cranks at 120°.	Two Cranks at 90°.	Two Cranks at 180°.	Three Cranks at 120°.
Case	I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XII	XIII	XIV
Phase Displacement ϕ ...	2°	2°	2°	2°	2°	2°	2°	2°	2°	2°	2°	2°	2°	2°
Angular Displacement for Bipolar Alternator α ...	1°	1°	1°	1°	1°	1°	1°	1°	1°	1°	1°	1°	1°	1°
Periodicity per second ...	50	50	50	50	50	50	50	50	50	50	50	50	50	50
Output, kilowatts	300	300	300	300	300	300	300	300	300	300	300	300	300	300
Cylinder Diameter, inches ...	35½	25¼	21⅛	25¼	18⅛	15¼	28¾	20½	17¼	...	...	...	...	...
Stroke, inches	18	15	13	15	12	10	36	30	24	...	...	...	...	...
I.H.P.	481	481	481	481	481	481	481	481	481	481	481	481	481	481
Revolutions per minute ...	230	273	300	273	333	375	88¼	103½	120	375	300	103½	103½	120
Number of Poles	26	22	20	22	18	16	68	58	50	16	20	58	58	50
Duration of Biggest Impulse	200°	110°	65°	95°	53°	53°	90°	53°	50°	70°	87°	54°	102°	70°
Coefficient of Fluctuation of Energy k	1.297	0.134	0.064	0.143	0.132	0.011	0.148	0.039	0.009	0.033	0.022	0.044	0.121	0.025
Reference to Diagram Number	1301	1302	1303	1305	1305	1306	1308	1309	1310	1311	1315	1316	1317	1318
Engine Energy per Revolution, E_0 ft.-lbs.	69,000	58,100	52,800	58,100	47,600	42,300	179,800	153,300	132,000	42,300	52,800	153,300	153,300	132,000
Limit Value of Fluctuation of Speed q	$\frac{1}{650}$	$\frac{1}{302.5}$	$\frac{1}{162.5}$	$\frac{1}{261}$	$\frac{1}{120}$	$\frac{1}{106}$	$\frac{1}{765}$	$\frac{1}{384}$	$\frac{1}{312}$	$\frac{1}{140}$	$\frac{1}{218}$	$\frac{1}{392}$	$\frac{1}{740}$	$\frac{1}{438}$
Necessary Energy in Equivalent Fly-wheel, E ft.-lbs....	29,085,225	1,177,481	274,543	1,084,194	376,980	24,645	10,178,325	1,147,776	175,328	97,720	126,658	1,322,020	6,863,130	722,700
Ratio, $E \div E_0$	422	20.3	5.2	18.6	7.9	0.58	56.6	7.48	1.33	2.3	2.4	8.9	44.7	5.5
Weight of Equivalent Fly-wheel at 80 ft. per second, tons	138	5.57	1.3	5.1	1.78	0.11	48.0	5.47	0.83	0.46	0.60	6.25	32.4	3.42

all cylinders. The crank-effort diagram is from the engine illustrated in fig. 1331.

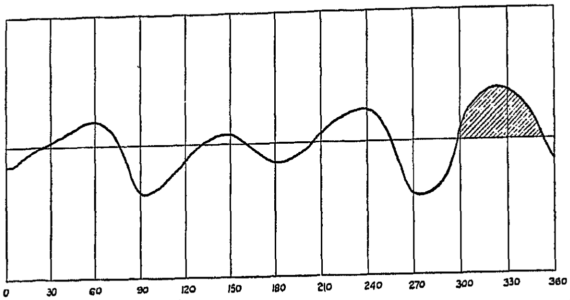


Fig. 1316, Case XII.—Crank-Effort Diagram. Double-Acting Cross-Compound Slow-Speed Engine—Two Cranks at 90°. Coefficient of fluctuation = 0.044

Cases XII and *XIII*.—Figs. 1316 and 1317 give the crank-effort diagrams, and call for no comment.

Engines are usually supplied with heavier fly-wheels than are indicated as necessary on Table IV, partly because the coefficient of fluctuation of energy at very light loads may exceed that at full load, but chiefly because during the act of switching an alternator into parallel with another there may be a momentary disturbance which might have awkward results.

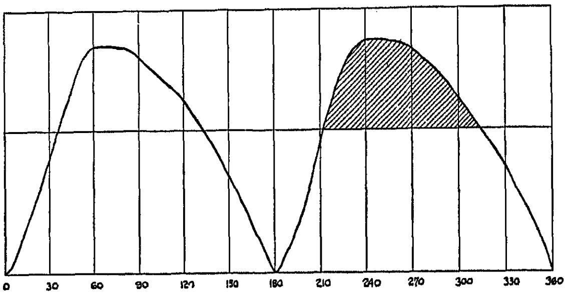


Fig. 1317, Case XIII.—Crank-Effort Diagram. Double-Acting Cross-Compound Slow-Speed Engine—Two Cranks at 180°. Coefficient of fluctuation = 0.121

Further, the momentary rise or fall in speed during a sudden large change of load must be kept within limits, and this is generally a far harder thing to accomplish than any reasonable degree of regularity of turning.

From the foregoing it is evident that, regarded from the point of view of keeping to a given allowable phase displacement between two alternators in parallel, the high-speed type of engine (whether single or double acting) has the advantage over slow-speed types, not only because a given limit of fluctuation of speed requires less fly-wheel energy and weight, but also because a greater fluctuation of speed is permissible owing to the lesser number of poles on the magnets

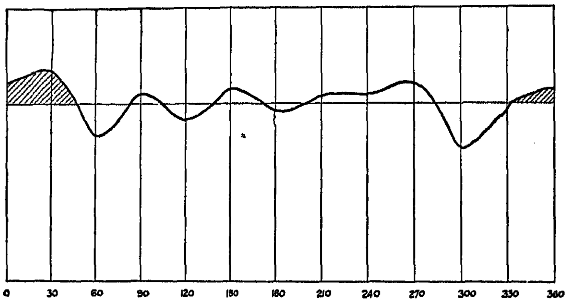


Fig. 1318, Case XIV.—Crank-Effort Diagram. Double-Acting Three-Crank Side-by-Side Triple-Expansion Slow-Speed Engine—Cranks at 120°. Coefficient of fluctuation = 0.025

CHAPTER IV

RECIPROCATING STEAM-ENGINES—STEAM CONSUMPTION

Theoretical Steam Consumption.—Since the elementary theory of the steam-engine has been gone through in a great number of text-books on the subject, it is not intended to do so again here. It is sufficient for the work in hand to recall to mind that, steam admitted into a cylinder and expanding in it while driving the piston before it, doing work, would, if the cylinder and piston were made of non-conducting materials, expand according to the adiabatic law.

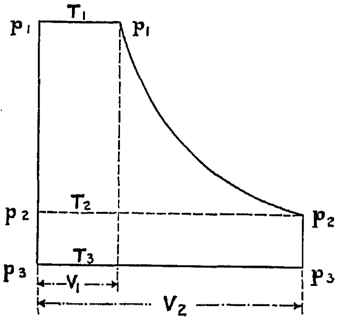


Fig. 1319

Let T_1 be the temperature (absolute) during the admission of steam to the cylinder.

T_2 the temperature reached at the end of expansion, just when the exhaust-port opens.

T_3 the temperature corresponding to the back-pressure p_3, against which the steam is expelled from the cylinder.

L_1 the latent heat (thermal units) of 1 lb. of saturated steam at temperature T_1.

Then, the thermal units due from 1 lb. of steam expanded from T_1 to T_2 and exhausted against a back-pressure corresponding to a temperature of T_3, is given by the equation—

$$U = L_1 \frac{T_1 - T_2}{T_1} + (T_1 - T_2) - T_2 \log_e \frac{T_1}{T_2} + \frac{144\,(p_2 - p_3)\,V_2}{778}.$$

Here p_2 is the terminal pressure (pounds per square inch) at end of expansion.

p_3 is the exhaust pressure.

V_2 is the volume (cubic feet) of the steam at pressure p_2.

The thermal units due, per pound of steam, from the portion $p_1 p_1 p_2 p_2$ of the diagram given, fig. 1319, are $L_1 \frac{T_1 - T_2}{T_1} + (T_1 - T_2) - T_2 \log_e \frac{T_1}{T_2}$.

The thermal units due, per pound of steam, from the portion $p_2 p_2 p_3 p_3$ are $\frac{144\,(p_2 - p_3)\,V_2}{778}$.

One indicated horse-power $= \frac{33000 \times 60}{778} = 2545$ thermal units per hour.

The mean effective pressure exerted by the steam during the stroke on each square inch of piston area is given by the formula—

$$\text{MEP} = \frac{\left(L_1 \frac{T_1 - T_2}{T_1} + (T_1 - T_2) - T_2 \log_e \frac{T_1}{T_2}\right) 778}{V_2 \times 144} + (p_2 - p_3).$$

In the above equation L_1 can be found from any published table of the properties of steam, or by calculation from $L = 1114 - 0.7\ T$. Only V_2 is not known. This, however, can be found by calculating the dryness fraction of the steam after adiabatic expansion from T_1 to T_2 from the equation—

$$\text{Dryness fraction} = x = \frac{\log_e \dfrac{T_1}{T_2} + \dfrac{L_1}{T_1}}{\dfrac{L_2}{T_2}}$$

L_2 is the latent heat of the steam at temperature T_2 found in the steam tables, or by calculation.

The tables further give the volume of a pound of saturated steam at all pressures, so that all that it is necessary to do is to multiply the volume of 1 lb. of saturated steam at temperature T_2 by the "dryness fraction" x, as found from the above, and V_2 is obtained. It is because steam expanding in a non-conducting cylinder, and doing work while so expanding, parts with some of its energy, that it becomes wet; the balance of dry steam left occupies a proportionately less volume.

Being now provided with all data for finding the thermal units due from one pound of steam expanded between given limits of temperature, and exhausted against another temperature, it is a short step to find the number of pounds of steam per I.H.P. per hour theoretically required by the engine. For instance, if the calculation for thermal units due from *one* pound of steam works out at 254.5, obviously the theoretical consumption will be $2545 \div 254.5 = 10$ lbs. per I.H.P. per hour, since *one* I.H.P. = 2545 thermal units per hour. By working out a number of cases, such as the theoretical consumption with 10, 20, 30, 40, &c. lbs. initial pressure, with 2, 4, 6, 8, 10, &c. expansions, and back pressure in all cases zero, also the theoretical mean pressures for the same conditions, it is possible to draw a series of curves which will show how the theoretical consumption of an engine varies according to the conditions of working.

In fig. 1320 the base-line is divided up into a number of equal spaces, which are taken to represent mean pressures, referred to the large piston of the engine. These divisions will also represent the indicated horse-power of the engine for the case when 1 lb. mean pressure gives 1 I.H.P. The theoretical consumption in pounds per I.H.P. hour is multiplied by the theoretical mean pressure, thus giving the total consumption of the engine in pounds per hour, and the result is plotted above the corresponding I.H.P.

Thus—For 110 lbs. absolute initial pressure,
0 " " back "
and 4 expansions,

the theoretical consumption will be found, from the formula, to be 13.9 lbs. per I.H.P. per hour, and the theoretical mean pressure is 62.2 lbs. per square inch. Multiplying 13.9 by 62.2 gives 864.6 lbs. per hour.

Set off a distance vertically from the base, above 62.2 mean pressure, to represent 864.6 lbs. to any convenient scale. Complete the calculations for all pressures 10, 20, 30, &c., and 4 expansions, and plot the results as described. Similarly, for 6, 8, 10, &c. expansions, complete the calculations and the plotting. Next draw fair lines passing through all the calculated points for 4 expansions, 6 expansions, and so on. Finally, draw fair curves through all the points denoting consumptions at various

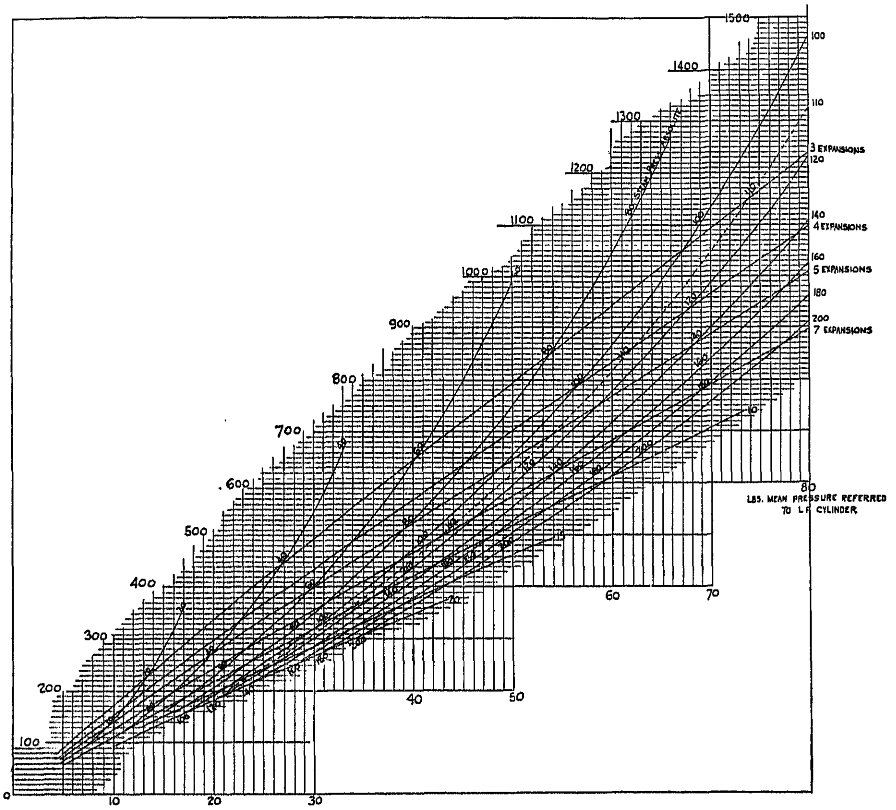


Fig. 1320.—Theoretical Standard-Consumption Engine. Curves of Consumption per Hour. Standard Engine gives 1 H.P. for 1 lb. mean pressure referred to L.P. piston

expansions, but with the same initial pressure, as, for instance, the curve 110, 110, 110,......, and the figure is complete.

A very important point to notice is, that the steam-consumption lines for *constant expansions* are very nearly *straight lines.* Notice also, that the calculated points for each 10 lbs. of initial pressure fall at *equal intervals* along the consumption lines, thus emphasizing the fact, that equal increments of initial pressure produce equal increments of mean effective pressure, provided the number of expansions of the steam remains unchanged.

A perfect engine, if governed by the "throttling" method, and having the cut-off fixed so as to give 4 expansions, would consume steam as shown by the nearly straight line marked "4 expansions". If, further, the initial pressure available was not more than 110 lbs. per square inch, 62.2 I.H.P. would be the maximum power of the engine with zero back pressure. If, however, the engine were governed by the "expansion" method, the pressure being maintained at 110 lbs. per square inch at all loads, and the expansion gear altering the cut-off as the load on the engine varied, it would consume steam as shown by the curve marked 110 lbs. initial pressure. The diagram may be drawn to a large scale, so as to contain a great number of lines and thus make it possible to read the consumption, direct from the curves, for any conditions of steam pressure and expansions; it must be remembered, however, that the consumptions shown by the curves are for steam exhausted against zero back pressure.

Supposing, now, it is desired to know the theoretical consumption for 110 lbs. initial pressure and 4 expansions, but with a back pressure of 14.7 lbs. per square inch; instead of dividing the 864.6 lbs. per hour by 62.2 lbs. mean pressure, it is necessary to divide by 47.5 lbs. mean pressure. Referring to fig. 1321 it will be seen that a strip of the theoretical indicator diagram, 14.7 lbs. high, is cut off, owing to the back pressure against which the engine has to expel the steam. As the mean pressure against 0 back pressure is 62.2 lbs. per square inch, so the mean pressure obtainable from the steam exhausted against 14.7 lbs. back pressure is 62.2 − 14.7 = 47.5 lbs. per square inch. The engine, however, takes in the same quantity of steam per hour as before, hence the consumption per I.H.P. per hour is 864.6 ÷ 47.5 = 18.2 lbs., and the maximum power of the engine only 47.5 I.H.P. Similarly, for any other back-pressure limit imposed upon the engine, for which the theoretical consumption has to be determined, the back pressure must be deducted from the mean pressure of the steam exhausted against 0 back pressure.

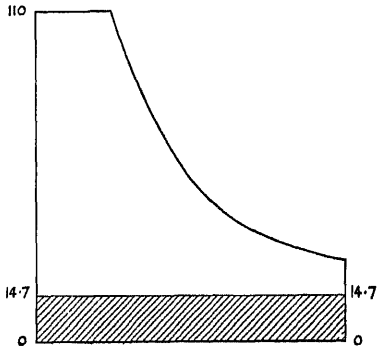


Fig. 1321

Of course no engine is perfect, and none made of non-conducting materials, consequently no engine has realized the steam consumptions shown as theoretically possible by fig. 1321. There are many imperfections of workmanship and design which give rise to leakage of steam past the valves especially, but also past the piston rings and so forth, and there is also condensation in the cylinder. Owing, also, to imperfections of valve gearing, and the need for having clearance between pistons and cylinder covers, as well as the unavoidable clearance in steam-ports, certain portions of the theoretical indicator diagrams are not obtained in practice; consequently, the mean effective pressure and indicated horse-power of the engine fall short of the theoretical mean pressure and horse-power.

Diagram Factor.—The above is the name given to the ratio the mean effective pressure, as measured from the indicator diagrams, bears to the theoretical mean effective pressure due from the steam, under the given conditions of pressure and expansions. It has been explained how the theoretical mean effective pressure exerted by the steam may be calculated, when the initial pressure, the number of expansions, and the back pressure are known; the reason for adopting the adiabatic expansion law for non-jacketed engines has also been given. As, however, the calculation of the probable diagram factor for a given engine is purely a matter of percentage, the use of one law of expansion rather than another cannot materially affect the result. For simplicity, therefore, the hyperbolic law of expansion (P V = constant) will be used in the following calculations. Quantitative examples will perhaps give the clearest demonstration of the reason why the theoretical mean pressure is not obtained in actual steam-engines.

Take the case of steam expanded, hyperbolically, from 100 lbs. absolute pressure per square inch, 4 times, and exhausted against 16 lbs. absolute back pressure. The mean effective pressure is 44.6 lbs. per square inch of piston area (fig. 1322, area *a b c d e a*), and this pressure is only obtainable in a perfect cylinder without clearance. If the cylinder has a clearance space equal in volume to 10 per cent of the total piston displacement, the steam will expand from volume 0.25 + 0.1 = 0.35, to volume 1.0 + 0.1 = 1.1, or $\frac{1.1}{0.35}$ = 3.14 times instead of 4 times. In order, however, that the same volume of steam may be taken into the cylinder per stroke as in the case fig. 1322, the valve motion must be so arranged that steam, on the return stroke of the piston, is compressed to the initial pressure (100 lbs.) just as the crank passes the dead-centre (fig. 1323). Measurement of the area of the diagram *a b g d h a*, and division by the length of the diagram *e d*, gives the mean height, which when multiplied by the scale of pressures will be found to be only 39.6 lbs. per square inch. Similarly, if the cylinder has 20 per cent clearance, the mean effective pressure will be found to be still less, viz. 33.5 lbs. per square inch, for the same expansions (fig. 1324).

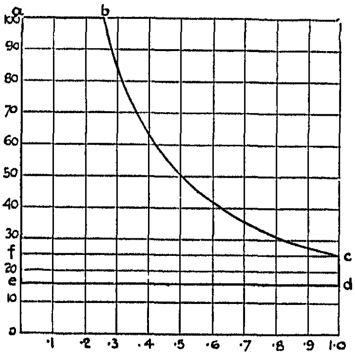


Fig. 1322

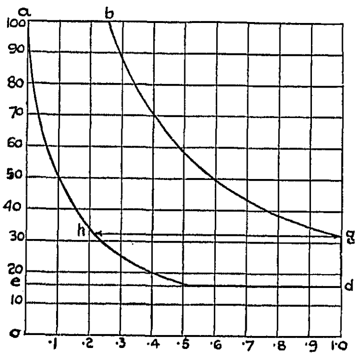


Fig. 1323

The differences 44.6 – 39.6 = 5 lbs. mean effective pressure,
and 44.6 – 33.5 = 11.1 „ „

are the losses due to 10 per cent and 20 per cent cylinder clearance respectively.

Beyond the loss due to clearance, there is a further loss due to imperfections of valve motion, whereby the corners of the indicator diagrams are rounded off owing to the ports not being *suddenly* opened to their full extent at release, and closed suddenly at cut-off and compression. Neglecting at present the corners so lost, it will be seen that the mean pressure obtainable with 10 per cent clearance in the cylinder is about 89 per cent of that obtainable in a cylinder without clearance.

Putting these figures in tabulated form, the diagram factors are as below:—

For 4 expansions, 100 lbs. absolute pressure, 16 lbs. absolute back pressure.

For 0	clearance,	mean effective	pressure	= 44.6	diagram	factor	= 1.00
,, 10 %	,,	,,	,,	= 39.6	,,	,,	= 0.89
,, 20 %	,,	,,	,,	= 33.5	,,	,,	= 0.75

The diagram factor given above is only concerned with the loss due to clearance; but its magnitude is also affected by all other losses of area in the card, such as falling admission line, rounded corners, &c.

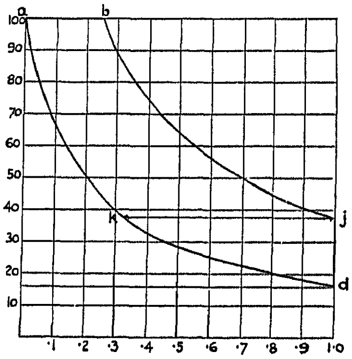


Fig. 1324

As an example of the effect the diagram factor has on the steam consumption of an engine, take the case of an engine with 165 lbs. absolute steam-chest pressure, 10 "nominal" expansions, and a back pressure of 1.5 lb. absolute. Calculation, or inspection of a large-scale diagram like fig. 1320, shows that the theoretical mean effective pressure under these conditions is 44.6 lbs. per square inch (about), and that the theoretical steam consumption is 10.5 lbs. per I.H.P. per hour (about).

Actual indicator diagrams from a Willans compound transfer engine with 10 nominal expansions, 164.6 lbs. steam-chest pressure, and 1½ lb. absolute back pressure, showed a mean effective pressure, referred to the L.P. pistons, of 38.7 lbs. per square inch. In this case, therefore, the "diagram factor" was 87 per cent. The theoretical steam consumption being 10.5 lbs. per I.H.P. per hour, and the "diagram factor" 87 per cent, the actual steam consumption could not possibly have been less than 10.5 ÷ 0.87 = 12.1 lbs. per I.H.P. hour, and it still remains to be shown why it is that the steam consumption was not even as low as this figure. It was, as a matter of fact, 14.8 lbs. per I.H.P. per hour.

The Missing Quantity, Condensation and Leakage.—If the indicator diagrams from any engine working with saturated steam be measured up, and the quantity of steam passing through the engine be calculated from them in the usual way, it is invariably the case that the quantity of steam so found is less than the actual steam consumption of the engine, often by a very considerable amount. Mr. Willans called the difference between the actual total steam used by the engine, and the steam shown by the indicator to have been present in the cylinder as steam, the "missing quantity". Experiments have been conducted by many eminent engineers in this and other countries to find what this "missing quantity" is due to, in order that it may be reduced and so increase the economy of the steam-engine.

The explanation generally given is that when the high-pressure steam enters the cylinder it finds the cylinder walls, piston, &c., cooled by having been exposed to the low-pressure exhaust steam. Some of the steam is condensed, and as the piston advances more and more of the cooled cylinder walls are exposed to the steam, and consequently condensation continues. This condensation continues until a stage is reached when the condensation on the newly exposed metal balances the re-evaporation of the liquid layer produced earlier in the stroke. Thus a film of water is produced upon the cylinder walls. This film of water is of course useless, and it is seen that more steam than is expanded must be sent into the cylinder.

The walls and piston are supposed to be reduced to the temperature of the exhaust just before admission, so that on this assumption initial condensation would be responsible for a large amount of steam. Many experiments have been carried out in order to determine the effect of varying circumstances on this "initial condensation". The influences of speed, size, ratio of expansion, steam jacketing, and superheating have each been considered, and expressions have been derived from the experimental results giving the initial condensation in terms of these variables.

Let us now briefly consider what would be the effect of these different circumstances. In the first place, the cylinder walls have just been cooled by exposure to the temperature of the preceding exhaust stroke when the incoming steam strikes them; consequently the steam parts with some of its heat and deposits water on the surfaces.

Secondly, radiation of heat from those parts of the cylinder and cover which, for practical reasons, cannot be covered with non-conducting material, tends further to reduce the temperature of the metal. This leakage of heat by radiation takes place at the expense of the heat of the incoming steam.

Thirdly, the design of the cylinder affects the missing quantity. For instance, in two cylinders of exactly the same capacity, and having the same percentage of cylinder clearance, the clearance may in the one case be enveloped in a larger amount of surface than in the other. The relation between the surface (S) and volume (V) of steam enveloped by that surface affects the missing quantity. Mr. Bodmer found that—

$$\frac{\text{The missing quantity}}{\text{Indicated steam at cut-off}} \propto \frac{\text{Clearance surface}}{\text{Volume behind piston}}.$$

Fourthly, imperfect drainage of water from the cylinder increases the missing quantity.

Fifthly, the range of temperature to which the cylinder surfaces are subjected affects the amount of condensation. The lower the temperature of the exhaust steam the greater the difference in temperature between that of the incoming steam and of the cylinder walls which it strikes, consequently the greater the initial condensation.

Sixthly, compression of the steam into the clearance space of the cylinder before admixture of the next charge of steam tends to lessen the difference of temperature between the incoming steam and the cylinder walls.

Seventhly, the density of the steam is another factor affecting condensation; *cæteris paribus*, the denser the incoming steam the less the *percentage* of the incoming steam condensed. Although the total amount of steam condensed is greater with greater density, yet there being more pounds of steam in a given volume, the surface enveloping these pounds of steam is less *per pound*, and consequently the *percentage* of incoming steam condensed is less.

Eighthly, the number of revolutions per minute made by the engine also affects condensation, as it determines the time during which each charge of steam is in contact with the cylinder wall. Some experimenters have found that the missing quantity varies *inversely as the square root* of the revolutions, others as the two-third power of the revolutions.

Willans speed trials show that the relation,

$$\frac{\text{Missing quantity}}{\text{Indicated steam at cut-off}} \propto \frac{1}{\sqrt{N}},$$

if not actually true, is at all events very close to the truth.

This does not embrace all the variables affecting condensation; it would be difficult to take account of the drainage of water from the cylinder, for instance, and the amount of compression in the cylinder. All that can be said is, that if the missing quantity in a certain engine is known, then the condensation in another engine may be predicted roughly, but only very roughly.

In commenting on the missing quantity above, the generally accepted view has been given that the missing quantity is due to some of the steam entering the engine being condensed on the inside surface of the cylinder, which has been cooled by contact with the exhaust steam.

We must now devote ourselves to a consideration of the later investigations on the subject, in which an endeavour was made to separate the condensation taking place from the losses due to leakage.

It is evident that the temperature of the walls on which the steam is condensed will affect the quantity that is condensed. If the wall temperature is high there will be but a small difference of temperature between the walls and the entering steam, and consequently but little initial condensation; whereas if the wall temperature be low then the temperature difference will be great, and consequently there will be a large amount of condensation.

It was pointed out by Professor Cotterill in his text-book on the *Steam Engine* that if he calculated the amount of steam condensed, assuming that the temperature varied from that of the admission steam to that of the exhaust, he got a figure about twice as great as the total missing quantity in even the most economical engines. The only conclusion that can be drawn from this is that the cylinder walls have a much smaller temperature range than the steam. Most modern research on the question of the missing quantity has been concerned with the determination of the wall temperatures, in order that the amount of condensation taking place can be assessed.

Mr. Bryan Donkin endeavoured to find the metal temperature range by drilling holes in an engine cylinder and inserting specially made mercurial thermometers. He could not observe the fluctuations of temperature with any degree of accuracy, owing to the thermometers not showing the changes instantaneously. He observed the important fact that the average temperature of the metal is higher than that of the steam. The next series of experiments to be considered are those of Professor Hall of Harvard University, who tried a thermo-electric method to get the surface range of temperature.

Another series of experiments were by Professors Callendar and Nicolson at Montreal (*Proceedings Institution of Civil Engineers*, vol. cxxxi). They employed the thermo-electric method suggested by Hall. The thermo-junctions in this case were made of the metal of the cylinder itself and wrought-iron wire. The electric circuit was made and broken at any time in the revolution by means of a contact maker fitted to the shaft. The temperature of the steam itself was also measured by means of a modification of Professor Callendar's well-known platinum thermometer, in which the temperature is derived from the changes in the electrical resistance of a piece of platinum wire.

These experiments showed clearly that although the steam temperature varied greatly, the wall temperature only varied some few degrees. In some experiments, when the steam temperature varied about 120° F., the wall temperature only varied about 3° or 4° F. It is therefore seen that if the walls have a change in temperature of so small an amount, that the condensation will be limited in amount and only a small proportion of the total missing quantity. We have therefore to seek elsewhere to find a solution to our problem, and these experiments of Messrs. Callendar and Nicolson showed that the greater part of this missing quantity was due to steam leakage past the valves—a possibility previously not considered. As a check, the leakage of the valve and piston was determined as nearly as possible under the conditions of running. This leakage was found to be proportional to the difference of pressure, and nearly independent of the speed through a considerable range.

The usual test for leakage with the valve stationary was found to be of little or no value. From a comparison of leakage tests, it was inferred that a valve in motion, however well fitted, was subject to leakage of a definite type. The leakage took place chiefly in the form of water by condensation and re-evaporation on the moving surfaces, and was directly proportional to the perimeter of the ports and inversely to the width of the bearing surface of the valve.

The next series of experiments to which attention is drawn are those of Professor Mellanby showing the effect of steam-jacketing upon the efficiency of a steam-engine (*Proceedings the Institution of Mechanical Engineers*, 1905). These experiments corroborate the work of Callendar and Nicolson as to the part played by initial condensation in the missing quantity.

In the summary of results at the end of the paper Dr. Mellanby states: "The jackets have little effect on the high-pressure, but have a considerable effect on the low-pressure cylinder upon initial condensation."

"The temperature cycles of the cylinder walls must be considerably less than that of the steam, (*a*) because the actual missing quantity is much less than it would have been had the steam and metal gone through the same temperature changes; (*b*) because the mean temperature of the metal is higher than that of the steam."

From these later experiments it is seen that the old assumption that the amount of condensation is unlimited is false, and show that the initial condensation is both limited and measurable, and that the greater part of the missing quantity has leaked past the valve in the form of water.

It is now intended to take the case of a triple-expansion engine by Hick Hargreaves and work out the theoretical consumption on the lines explained, and to compare this with the actual ascertained consumption of the engine to show the magnitude of the missing quantity.

Theoretical and Actual Steam Consumption.

Particulars of Engine—

H.P. cylinder diameter, 19 inches;
I.P. „ 29 „
L.P. „ 46 „ } stroke, 48 inches.

Three cranks. One cylinder operates each crank.

Actual piston area, 289 square inches H.P.
667 „ I.P.
1673 „ L.P.

Clearance volume, H.P. 0.441 cubic foot.
I.P. 0.75 „
L.P. 1.78 „

Steam-chest pressure, 175 lbs. per square inch; temperature, 370° F.

Vacuum, 26.6 inches = 1.6 lb. per square inch pressure; temperature, 118° F.

Nominal expansions, 15.7.

Revolutions per minute, $82\frac{1}{4}$.

Total mean effective pressure (referred to L.P. piston) = 26.6.

I.H.P. = 884.

Actual steam consumption, 12.6 lbs. per hour per I.H.P.

Let us now go through the calculation of the theoretical amount of steam needed per I.H.P. per hour.

Calculation of Theoretical Consumption—

$$V_1 = 2.57;\ V_2 = 40.35;\ p_1 = 175;\ p_2 = 7.7;\ p_3 = 1.6$$

$$T_1 = 370 + 461 = 831 \quad T_1 - T_2 = 189 \quad \frac{L_1}{T_1} = 1.025$$

$$T_2 = 181 + 461 = 642$$

$$\frac{T_1}{T_2} = 1.294 \quad \log_e \frac{T_1}{T_2} = 0.2577$$

Thermal units due from 1 lb. of steam

$$U = \left(\frac{L_1}{T_1} + 1\right)(T_1 - T_2) - T_2 \log_e \frac{T_1}{T_2} + \frac{144\,(p_2 - p_3)\,V_2}{778}$$

$$= 2.025 \times 189 - 642 \times 0.2577 + \frac{144\,(7.7 - 1.6)\,40.35}{778}$$

$$= 382.7 - 165.5 + 45.5$$

$$= 262.7 \text{ British thermal units.}$$

But 1 I.H.P. is equivalent to $\frac{33000 \times 60}{778} = 2545$ British thermal units per hour,

Therefore the theoretical consumption is given by $\frac{2545}{262.7} = 9.7$ lbs. of steam per I.H.P. per hour.

This is the result for an engine which draws out a card showing no diagram losses, that is to say, a case where the diagram factor is unity. The next thing we should attempt is the determination of the diagram factor. From what has been said previously, we know that the diagram factor is the ratio of the mean pressure obtained from the cards to that theoretically due from steam working under the specified conditions.

In fig. 1325 the indicator cards for the three cylinders have been combined and drawn out on the one diagram—an explanation of the method adopted in drawing out the cards is given later, and from which we can get the mean pressures exerted by the steam in the cylinder, and also that which would be obtained if the theoretical card could be obtained.

From this expanded diagram let us take the data necessary to calculate the diagram factor.

First let us calculate the mean pressure theoretically obtainable under the conditions specified, then we will consider the setting out of the cards on the diagram, and finally will obtain the mean effective pressure that is given by the cards. From the well-known formula for hyperbolic expansion the mean pressure due from the steam is

$$p_m = p_1 \cdot \frac{1 + \log_e \gamma}{\gamma} - p_3,$$

where p_m is the mean effective pressure, p_1 the initial pressure, and p_3 the exhaust pressure, and γ is the ratio of expansion.

$$\therefore\ p_m = 175 \cdot \frac{1 + 2.7535}{15.7} - 1.6$$

$$= 40.2 \text{ lbs. per square inch.}$$

Calculation of Diagram Factor.—Set off the volume of L.P. cylinder, 46.50 cubic feet, on the base-line. Let pressures be plotted to scale vertically. For 15.7 nominal expansions the H.P. cut-off volume will be 2.96. Draw the hyperbolic curve *a b c d* (fig. 1325), representing the expansion of steam from 175 lbs. pressure at 2.96 cubic feet volume to 46.5 cubic feet volume. Draw the line *e f* at 1.6 lb. absolute back pressure. The area *a b c d e f g* represents the theoretical diagram for hyperbolic expansion from 175 lbs. absolute pressure, 15.7 expansions, and exhausted against 1.6 lb. absolute back pressure.

From *g* set back *g h* = 0.441 cubic foot, representing the H.P. cylinder clearance. Erect perpendiculars at 8.04 cubic feet and 18.55 cubic feet, the H.P. and I.P. cylinder volumes respectively, and through their points of intersection with the hyperbolic curve draw the horizontals *b j*, *c k*, which show the division of pressures and consequently of temperature between the three cylinders. Set back from 0 volume line the I.P. clearance and L.P. clearance respectively, viz. 0.75 cubic foot and 1.78 cubic foot.

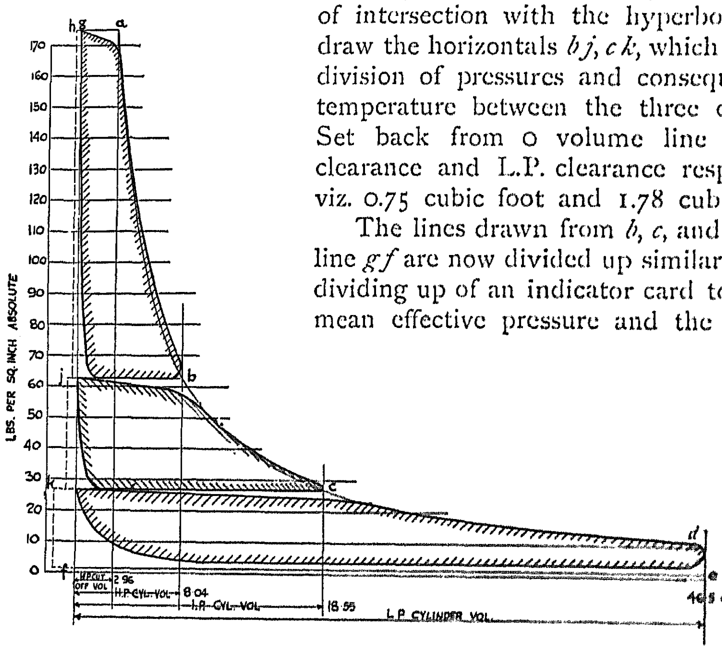


Fig. 1325

The lines drawn from *b*, *c*, and *e* to the line *g f* are now divided up similarly to the dividing up of an indicator card to find its mean effective pressure and the pressure for each of these ordinates—which are not shown on this diagram—is read off from the ordinary indicator diagram and set up on this diagram. By drawing through the many points so obtained the three indicator diagrams shown have been obtained. By measuring up these diagrams so drawn, it will be found that the mean effective pressures are as below:—

H.P. diagram M.E.P. = 64.2 lbs. per sq. in., or 10.8 referred to L.P.
I.P. " " = 21.8 " " 8.6 " "
L.P. " " = 15.5 " " 15.5 " "
Total M.E.P referred to L.P. = 34.9

A diagram factor of 34.9 ÷ 40.2 = 87 per cent may reasonably be expected from this engine. By reason, therefore, of the non-realization of the theoretical mean effective pressure, the steam consumption will be raised from 9.7 to 9.7 ÷ 0.87 = 11.1 lbs. of steam per I.H.P.

Thus the difference between the indicated and the actual steam consumption per I.H.P. is 12.6 − 11.1 = 1.5 lb. per I.H.P. hour, or 11.9 per cent of the actual steam consumption.

In these calculations for theoretical steam consumption and theoretical mean pressure the volumes are those occupied by one pound of steam, whereas in the diagram factor calculations the volumes were those of the actual cylinder of the engine under consideration.

Thus the actual L.P. cylinder contained 46.5 cubic feet of steam, or $\frac{46.5}{40.55}$ = 1.15 lb. of steam per stroke.

Effect of Throttle Governing and Automatic Expansion Governing upon Economy.—Believing that numerical examples often convey more meaning than general statements, it is intended to construct a diagram giving the steam consumption of an engine governed by the throttle method, and also of the same engine governed by the variable-expansion method, and show thereby what effect those two methods of governing have on the economy of working.

Take the case of a compound double-acting high-speed engine, with cylinders arranged tandem fashion on a single crank.

Engine dimensions—H.P. cylinder diameter = 9 inches.
L.P. „ = 18 „
Stroke ... ... = 10 „
Revolutions per minute, 375.

The figures are those obtained on trial for the various conditions specified, whereas in the diagram factors the volumes were those of the actual cylinder of the engine under consideration.

VARIABLE-EXPANSION GOVERNING (INITIAL PRESSURE 150 LBS. PER SQUARE INCH), ENGINE WORKING NON-CONDENSING

I.H.P.	Steam per Hour.	Steam per Hour per I.H.P.
23.3	1086	46.6
72.6	2025	27.9
104.5	2590	24.8
160.5	3835	23.9
194	4776	24.6

ENGINE WORKING CONDENSING (VACUUM 26 INCHES)

I.H.P.	Steam per Hour.	Steam per Hour per I.H.P.
25	715	28.8
69	1348	19.5
87	1650	19.0
119	2161	18.2
176	3156	17.9
198	3540	17.8

THROTTLE GOVERNING, NON-CONDENSING

Initial Pressure.	I.H.P.	Steam per Hour.	Steam per Hour per I.H.P.
150 lbs. per sq. in.	194	4800	24.8
130 " "	163	4090	25.0
110 " "	132	3400	25.7
90 " "	102	2700	26.4
70 " "	70	2005	28.6
50 " "	39	1310	33.6
30 " "	8	640	80

THROTTLE GOVERNING, CONDENSING (VACUUM 26 INCHES)

Initial pressure.	I.H.P.	Steam per Hour.	Steam per Hour per I.H.P.
150 lbs. per sq. in.	198	3580	18.1
130 " "	172	3110	18.2
110 " "	145	2705	18.6
90 " "	118	2200	18.7
70 " "	91	1750	19.3
50 " "	64	1310	20.5
30 " "	40	870	21.8

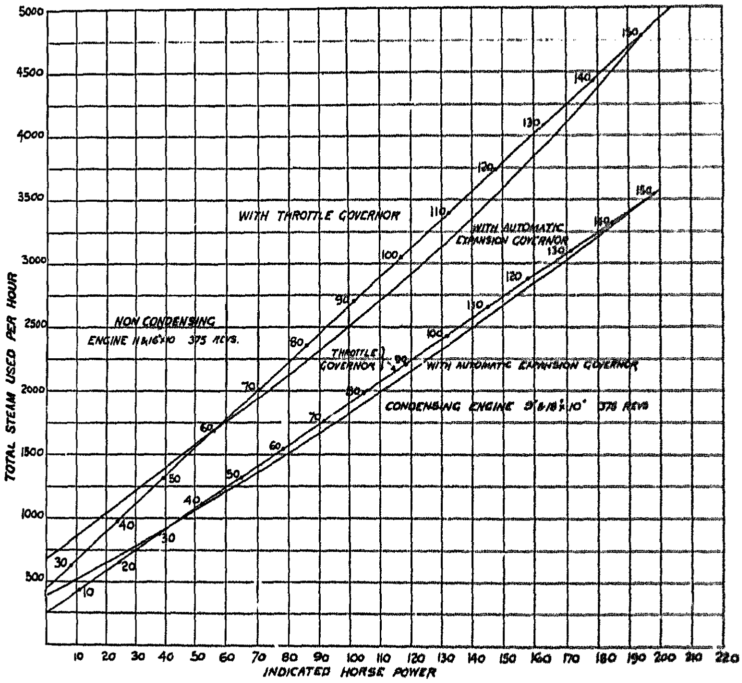


Fig. 1326.—Consumption (lbs. per hour) of Engine, with Automatic Expansion and with Throttle Governor. Compound Engines

A diagram (fig. 1326) can now be made showing how many pounds

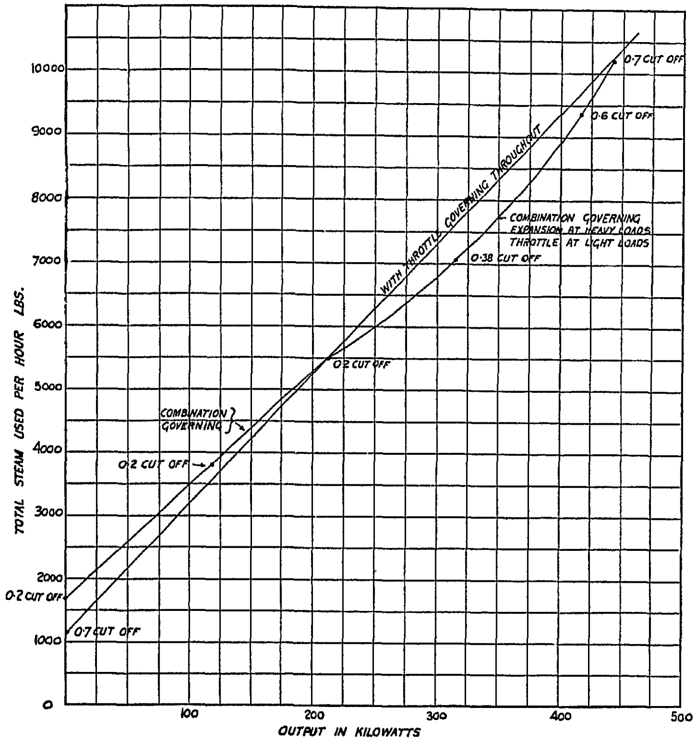


Fig. 1327.—Willans Three-Crank Triple-Expansion Engine and Parker Generator Condensing Steam-Pressure 153 lbs. per sq. inch

of steam per hour will be consumed by this engine at every load when governed by the expansion method and when governed by the throttling method.

A fair curve drawn through all the consumption points obtained with varying expansions and constant initial pressure (viz. 150 lbs. per square inch) shows the effect of expansion governing upon steam consumption.

In the case where the regulation is by means of different initial pressures and constant expansions (*i.e.* the throttling method) the steam consumption per hour plotted against the resulting indicated horse-power gives a straight-line graph; this straight line is known as the "Willans line".

For non-condensing working, fig. 1326 shows that the gain in economy obtained by expansion governing over throttle governing in this particular case at three-quarters load amounts to 6½ per cent, 6 per cent at half load, and 0 per cent at quarter load. For condensing engines the gain in economy with expansion governing is much less marked.

Fig. 1327 gives the actual steam consumption curves for a triple-expansion Willans engine, first with the expansion gear at work and afterwards with the cut-off fixed and the engine governed by throttle only.

On account of the heavy loss at light loads that would occur with the expansion method of governing, the makers of this engine have adopted the governor gear previously described, by which the expansion method is stopped at or about the load when further gain by expansion governing cannot be obtained, and throttle governing pure and simple is substituted for the lower loads. Fig. 1327 shows the effect on economy of this combination method of governing very well. In slow-speed engines the full advantage of the expansion method of governing is far more easily obtained than in high-speed engines, since it is possible to employ drop valves and Corliss valves with trip gear, which, so far, at all events, has not been successfully employed on high-speed engines.

Superheating.—There is no doubt that an engine using a certain quantity of saturated steam per hour will use a less weight of steam per hour, for the same horse-power, if supplied with superheated steam.

As an illustration of this economy we have the results of some experiments carried out by Professor Ripper upon a Schmidt single-cylinder engine.

Consumption per I.H.P. Hour.	Quality of Steam.	Gain by Superheat.
40	Saturated	
33·3	100° F. superheated	16.7
29·3	150° F. „	26.7
22.7	250° F. „	43·3

The saving in steam shown in these cases is due to the reduction of the "missing quantity". If engines could be constructed so that the indicated and actual weights of steam used were the same, there would be but few cases where the use of superheated steam would pay.

The gain in economy is not wholly due to a reduction in the initial condensation, or the metal walls would have their temperature range greatly reduced. The gain is due to the fact that wet steam will leak past a valve much more readily than dry steam, and that superheated steam cannot leak past so rapidly as dry steam, thus the valve leakage in the engine is greatly diminished. To illustrate the effect of high superheat on economy, the following results have been taken from a report of Mr. Michael Longridge on an engine specially designed for high superheat by the makers, Messrs. Cole, Marchent, and Morley of Bradford.

Engine dimensions—H.P. cylinder, 21 inches diameter.
L.P „ 36 „
and stroke, 36 „

	1	2	3	4	5	6
Pressure at engine (lbs. per square inch gauge)	117·5	117·5	117·5	117	114·5	114·5
Temperature of steam entering H.P. cylinder, degrees F.	601	590	569	580	558	550
Superheat, degrees F.	253	242	221	232	210	204
R.P.M.	100.6	100.7	100.6	100.7	100.7	100.7
I.H.P.	481	461	347	334	258	145
Steam per hour, lbs.	4379	4272	3088	2863	2240	1272
Steam per hour per I.H.P. ...	9·1	9·3	8.9	8.6	8.7	8.7

Consumption per B.H.P. and Works Cost.—A point which is often lost sight of in speaking of engine consumptions is that the real criterion of an engine performance is not the steam used per I.H.P. but *per B.H.P.* It matters little to the man who pays the bill whether he gets a good result per I.H.P. or not, so long as the cost of an *available* horse-power is low. It may easily happen that an engine showing a very low steam consumption per I.H.P. does not give an effective horse-power for any less steam than another engine with a less showy result per I.H.P.

Low steam consumption per I.H.P. may be obtained by expanding the steam nearly down to the pressure of the exhaust, but in order to do so it is necessary to have very big cylinders for the power required. The mean effective pressure is necessarily less with greater number of expansions, but the crank-shaft and connecting-rods have to be designed with regard to the initial pressure on the piston as well as the mean pressure; and consequently big cylinders generally lead to heavier shaft and rods. Under these circumstances the extra loss in friction of the bigger engine over that of the smaller engine may easily outweigh the gain in steam consumption per I.H.P., so that the net result per brake H.P. is actually worse. Two quite possible cases may be given to illustrate this:—

Engine A consumes 12 lbs. of steam per I.H.P., and wastes 13 per cent of its indicated power in friction, the result being that each brake horse-power requires 13.8 lbs. of steam for its production. Engine B consumes 12.7 lbs. of steam per I.H.P., but wastes only 8 per cent of power in friction, the result being that each brake horse-power in this case also requires 13.8 lbs. of steam for its production. Enough has been said to point out that low cost of an indicated horse-power does not necessarily mean low cost of an available horse-power; and that consequently the economy per brake horse-power is the best measure of an engine's performance, the economy per kilowatt being the best measure of a steam dynamo's performance. Further, it may be said that it is not *always* the best policy to choose the engine which is most economical of steam for a certain work. In places where coal is cheap, "first cost" of the engine may be of more importance than low steam consumption. Generally speaking, the most economical engines as regards steam consumption are the bigger engines, and cost more to build than smaller engines. The engineer has in all cases to do the best he can financially. If money can be borrowed at a small rate of interest, an engine possessing all the details making for economy in steam consumption may earn a good interest on the capital, and soon pay for its extra cost; if money, however, is dear, it is better to buy a smaller and cheaper engine, provided it will satisfactorily do its work, but which will probably consume a little more steam than the bigger and more costly engine.

The cost of steam, and consequently of coal, is not the only expense incurred in working; there is the cost of oil and stores, cost of attendance, repairs and maintenance, to be considered as well. In the electricity supply stations in London (taking one city only, so as to eliminate variations in price of coal) the average works costs of supplying a Board of Trade unit of electricity are as below:—

LARGE SLOW-SPEED ENGINES, BOILERS, AND AUXILIARY PLANT

	Coal.	Oil, Waste, Stores, &c.	Wages.	Repairs and Maintenance.	Works Costs.
Pence per unit ...	1.12	0.16	0.34	0.41	2.03
Portion of total costs	55.2%	7.9%	16.7%	20.2%	

SMALL HIGH-SPEED ENGINES, BOILERS, AND AUXILIARY PLANT

	Coal.	Oil, Waste, and Stores.	Wages.	Repairs and Maintenance.	Works Costs.
Pence per unit ...	0.91	0.09	0.27	0.32	1.59
Portion of total costs	57.2%	5.7%	17.0%	20.1%	

The stations use the same classes of boilers (Babcock, Paxman Economic, Loco type, Lancashire, and Marine) in fairly level proportions as between slow-speed and high-speed stations, so that the boiler upkeep and economy of steam production may well be considered as the same fixed amounts per unit of electricity generated. Also, the upkeep of the electrical portion of the plant is probably similar in all stations. However this may be, it is seen that the high-speed stations produce electricity considerably cheaper than the slow-speed stations. Probably this accounts for the enormous strides made of late years in the high-speed-engine business.

Correct Measure of Engine Performance.—A committee of members of the Institution of Civil Engineers met in 1896 to consider and report upon the subject of a standard of thermal efficiency for steam-engines. This committee, and Mr. Mair Rumley before them, pointed out that the expression of the thermal economy of a steam-engine in pounds of feed-water per hour is misleading, because the number of heat units required to produce a pound of steam depends upon the circumstances of its production. The error is not great when saturated steam is used, but when superheated steam is employed it may be very considerable. A pound of saturated steam at 50 lbs. pressure requires 1167 British thermal units for its production, and at 150 lbs. pressure absolute it requires 1191 British thermal units. One pound of saturated steam at 150 lbs. pressure, therefore, contains about 2 per cent more heat than 1 lb. at 50 lbs. absolute pressure. One pound of steam at 150 lbs. absolute pressure and 558.3° F. temperature contains about 1287 British thermal units, or 10 per cent more heat than the pound of saturated steam at 150 lbs. pressure. Thus it is seen that to compare the performance of an engine using superheated steam with that of one using saturated steam, by taking pounds of feed-water per hour as a reference, gives an inflated idea of the value of superheat.

The committee recommended, in comparing the performances of various engines:—

1. That "thermal efficiency", as applied to any heat engine, should

mean the ratio between the heat utilized as work on the piston by that engine, and the heat supplied to it.

2. That the heat utilized be obtained by measuring the indicator diagrams in the usual way.

3. That in the case of a steam-engine, the heat supplied be calculated as the total heat of the steam entering the engine, less the water-heat of the same weight of water at the temperature of the engine exhaust, both quantities being reckoned from 32° F.

4. That the temperature and pressure limits, both for saturated and superheated steam, be as follows:—Upper limit: The temperature and pressure close to, but on the boiler side of the engine stop-valve, except for the purpose of calculating the standard of comparison in cases where the stop-valve is purposely used for reducing the pressure. In such cases the temperature of the steam at the reduced pressure shall be substituted. In the case of saturated steam, the temperature corresponding to the pressure can be taken. Lower limit: The temperature in the exhaust-pipe close to, but outside the engine. The temperature corresponding to the pressure of the exhaust steam can be taken.

5. That a standard steam-engine of comparison be adopted, and that it be the ideal steam-engine working on the Rankine cycle between the same temperature and pressure limits as the actual engine to be compared.

6. That the ratio between the thermal efficiency of an actual engine and the thermal efficiency of the corresponding standard steam-engine of comparison be called the "efficiency ratio".

7. That it is desirable to state the thermal economy of a steam-engine in terms of the thermal units required per minute per indicated horse-power, and that, when possible, the thermal units required per minute per brake horse-power be also stated.

8. That, for scientific purposes, there be also stated the thermal units required per minute per horse-power by the standard engine of comparison.

It should be pointed out that the standard engine of comparison adopted is one which expands its steam right down to the temperature in the exhaust-pipe, whereas in the calculations for theoretical consumption just made, of course the actual number of expansions to be used in the engine have been taken. The reason for comparing the performance of an engine with a "standard engine of comparison" which expands its steam completely, is to give a comparison with the "best possible" that can be done between the limits of temperature at, but just outside, the engine. If an engine gives a big drop in pressure from that at the stop-valve to that in the cylinder it is a defect of the engine, unavoidable perhaps, but still a defect, and it would be inaccurate to alter the "standard of comparison" and thus make the engine appear more efficient than it is. Similarly, though it may be advisable to exhaust the steam before it has been completely expanded, it would be misleading if the standard of comparison were altered to suit. The "standard engine of comparison" is, therefore, rightly fixed as one making best use of the steam between the given limits of temperature *at* the engine.

CHAPTER V

RECIPROCATING STEAM-ENGINES—VARIOUS TYPES

In this chapter are given a few illustrations and descriptions of various types of engine used for dynamo driving. There are many engines in successful operation at the present day, but only a few typical examples can be given here, beginning with single-acting high-speed engines, going on to double-acting high-speed engines, and finally slow-speed engines.

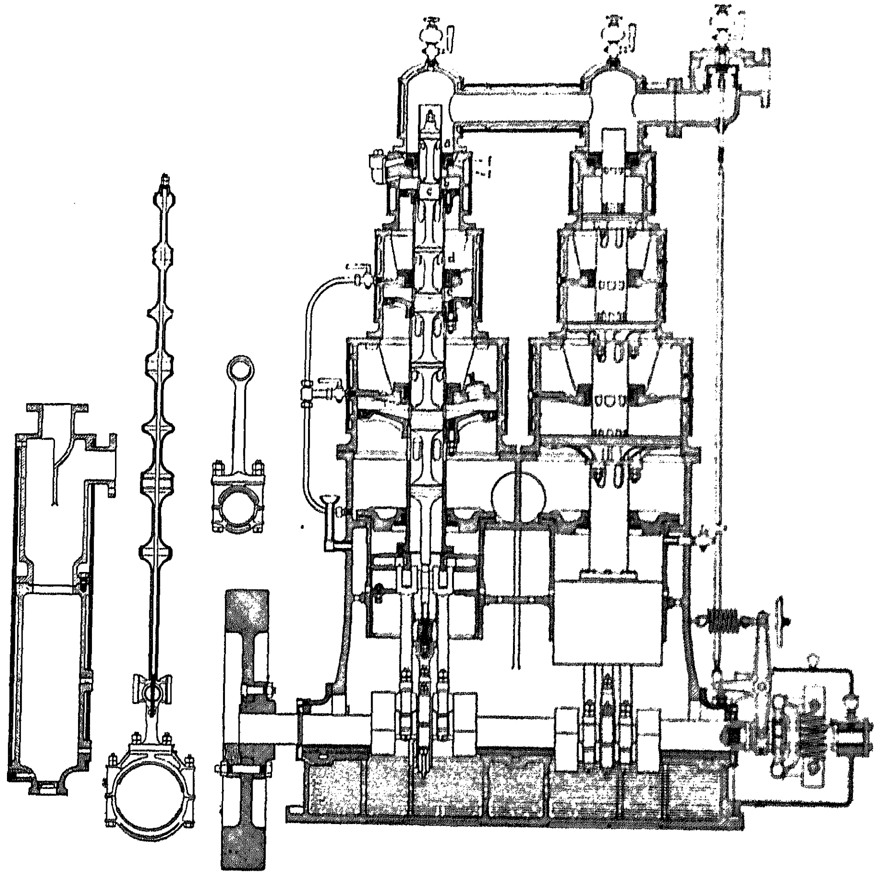


Fig. 1328.—Willans Central-Valve Engine

The Willans Engine.—This engine is single-acting, and is made with two or three cranks as may be required, each crank being driven by a complete engine, whether with a single cylinder, compound cylinders, or three cylinders. Fig. 1328 is a section of a triple-expansion two-crank engine. The down-stroke is the working or effective stroke: during the first portion of it the steam pressure on the piston presses the big end connecting-rod brass against the crank-pin, and the crank-shaft against the lower main bearing brass. On the up-stroke, when no work is being done, and when the steam is being merely exhausted from the several cylinders, the inertia of the parts as they undergo acceleration by the crank also keeps them in compression for the earlier portion of the stroke; in the latter portion they are cushioned by the arrangement to be presently described. The result is that, although the rate of turning is very high and the inertia correspondingly great, in no part of the stroke, up or down, do the parts tend to separate; consequently there is no knocking.

The engine is built with a complete engine on each crank in order to obtain even turning, as has been described in Chapter III, and although this entails small cylinders, their construction and the method of steam distribution is such that the clearances are unusually small for the sizes of cylinders employed. The method of steam distribution by a central valve (*i.e. central in the cylinder*) is directly responsible for the smallness of the cylinder clearances, and enables the engine to work as economically as the best types of slow-speed engine with large cylinders.

The cushioning to overcome the inertia of the parts during the later portion of the up-stroke is accomplished in the following way:—The cross-head guides take the form of bored cylinders, and the cross-heads are pistons. The top of the guide cylinder is closed, and on the up-stroke the air in it is compressed to the extent necessary to cushion the piston and other parts. The power stored in the air by compression on the up-stroke is given out again during the next down-stroke, without sensible loss. The reciprocating parts are thus held down at all times and do not depend for cushioning upon the compression of steam in the cylinders. The steam is distributed throughout by the hollow piston-rod. It enters from the steam-chest by the cut-off ports *a*. By the movement of the line of piston-valves, which work inside the piston-rod, driven by the eccentric, the steam passes into the high-pressure cylinder by the ports *b*. The ring of ports *b* is the only inlet and outlet from the cylinder, and moves up and down with the piston.

The valve gives just the same steam distribution as an ordinary slide-valve, with a slow cut-off at about three-quarters stroke. The actual cut-off is, however, effected by the ports *a*, which at a point in the stroke pre-arranged in the design of the engine, or controlled by the governor in automatic expansion engines, pass down through a ring-packed gland into the cylinder, and so leave the steam-chest, thus preventing the further supply of steam for that revolution. As the cut-off movement corresponds with that of the pistons themselves, the cut-off is very prompt, and shows a sharp corner on the diagram. After the steam has worked expansively in the high-pressure cylinder, the valve *c* passes above the ports *b* and opens communication from the working end of the cylinder to the space below the piston, which is called the receiver, but which also forms the steam-chest for the intermediate-pressure cylinder. During the up-stroke the steam is simply transferred from one side of the piston to the other; the whole cylinder, including the working end, at that time forms part of the receiver. When the next down-stroke commences, the steam in the first receiver is passed into the intermediate-pressure cylinder through the ports *d* and the ports *e*. Cut-off in this case is given by the ports *d* passing through the gland on the intermediate-pressure cylinder-head. The passage of steam into the low-pressure cylinder from the low-pressure steam-chest is effected in a similar manner.

In a compound engine there are but two cylinders in series; in a simple engine there is only one over each crank.

The cranks and all the working parts, except the cylinders and valves, are lubricated by the splash of the cranks in the crank chamber, where the lubricant consists of a mixture of oil and water. The points of cut-off in the various cylinders are adjusted in fixed expansion engines by means of the thickness of the rings *f* placed below the gland boxes. The method of varying the cut-off by means of the governor has been described in Chapter II.

Below are a few figures relative to the steam consumption and efficiencies of these engines:—

450 H.P. compound non-condensing engine at Whitechapel; Mather & Platt dynamo; 155 lbs. steam pressure; 30.48 lbs. per kilowatt-hour; 84 per cent combined efficiency.

700 H.P. compound non-condensing engine at Chelsea; 160 lbs. steam pressure; 29.7 lbs. per kilowatt-hour; 81½ per cent combined efficiency.

450 H.P. triple-expansion engine at Buenos Ayres; E.C.C. dynamo; 190 lbs. steam pressure; 27.5 lbs. per kilowatt-hour, non-condensing; 21.3 lbs. per kilowatt-hour, condensing.

1200 H.P. triple-expansion engine; Central Electric Oerlikon three-phase alternator; 190 lbs. steam pressure; 31.75 lbs. per kilowatt, non-condensing with by-passes open; 22.17 lbs. per kilowatt, condensing; combined efficiency, 79.7 per cent.

The Peache High-Speed Engine.—This engine also is single-acting, the balance of pressure being always towards the crank-shaft. It is of a special type, inasmuch as the low-pressure piston takes steam on its under side, thereby obviating the heavy grind over the bottom centre usual in single-acting types. The turning effort is, in consequence, much more even than in other types of single-acting engines. The high-pressure steam-chest consists of the annular space round the main body of the valves, and is situated partly in the high-pressure cylinder casting and partly in the low-pressure cylinder casting. Steam from the throttle-valve passes to a branch on the low-pressure cylinder casting, connected by passages in that casting to the three high-pressure steam-chests. This construction facilitates the removal of any high-pressure cylinder without breaking any joint but that between it and the low-pressure cylinder. Steam passes under the edge of the high-pressure valve to the top of the high-pressure piston, and is exhausted over the top of the high-pressure valve down through the centre of the main body of the valve, and thence over the top of the low-pressure valve to the *under side* of the low-pressure piston, and finally it is exhausted under the lower edge of the low-pressure valve to the exhaust chamber.

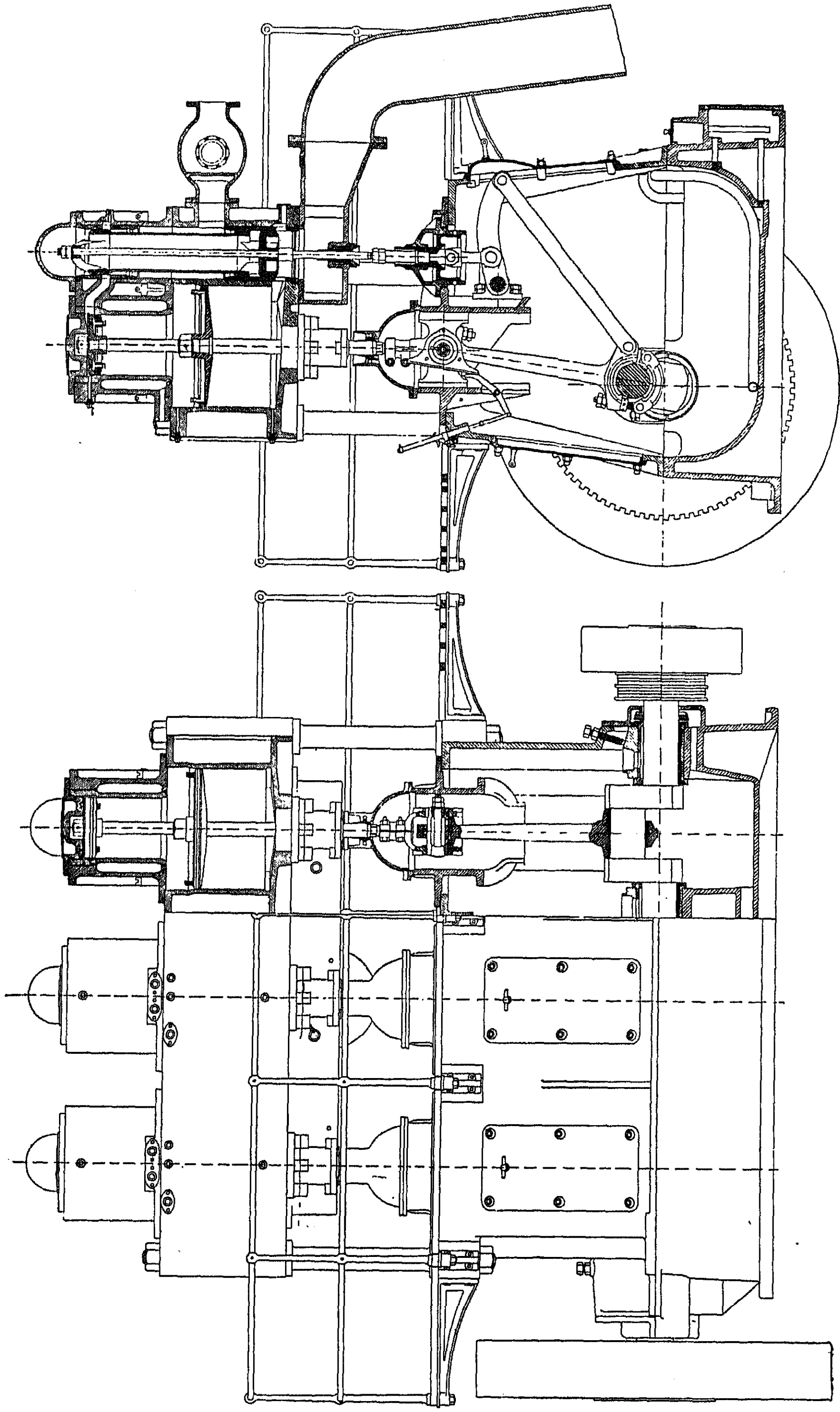


Fig. 1329.—Peache Compound Engine

In the bore of the high-pressure cylinder are cut two small by-pass ports, over which the high-pressure piston passes at the top of its stroke, thus placing the space between the high-pressure and low-pressure pistons in direct connection with the upper side of the high-pressure piston. This space, which is called the controlling cylinder, is thus at the top of the stroke filled with steam at approximately the same pressure as that in the high-pressure cylinder; on the down-stroke this steam expands, and on the up-stroke it is compressed, until at the top of the stroke it again reaches the pressure of the steam in the high-pressure cylinder, any small loss that may occur by leakage being made good by fresh steam passing from the high-pressure cylinder. The function of this controlling cylinder is to keep a slight excess of pressure in a downward direction throughout the up-stroke. Thus on the up-stroke the steam acts on the under side of the low-pressure piston, but instead of being allowed to act direct on the crank, it does work in compressing the steam in the controlling cylinder and in displacing the steam under receiver pressure from the high-pressure cylinder. The work thus done in compressing steam in the controlling cylinder is given out again by its expansion on the next down-stroke, and the work done in transferring steam under pressure from the high-pressure cylinder to the receiver is utilized in the low-pressure cylinder. The number of expansions of the steam in this controlling cylinder is entirely independent of the number of expansions of the working steam in the cylinder proper, since the opening from high-pressure cylinder to controlling cylinder is only during a very short portion of the stroke just at the turn of the top dead-centre. The high-pressure piston itself cuts off steam from the controlling cylinder long before the piston-valve cuts off steam from the cylinder. By providing more or less "clearance" in the controlling cylinder any desired ratio of expansion of the play steam in it may be obtained. The valves are always subject to receiver pressure, which tends to maintain them in constant thrust. A small air-buffer cylinder is also provided, fixed in the working chamber to supplement the "holding-down" power.

The crank-shaft is placed out of line with the cylinders. This gives a nearly straight connecting-rod during the downward working stroke, when the pressures are greatest, and keeps what pressure is due to the angle of the connecting-rod on the back cross-head slide. On the up-stroke, when the angle of the connecting-rod is greater, the downward pressures are only sufficient to balance the inertia forces of the reciprocating parts; and, therefore, although the angle of the connecting-rod is considerable, the pressure transmitted to the cross-head slide is small, and what there is is still on the *back* guide. Thus all chance of knock of the cross-head between the guides is prevented. The valves are worked from a point on the connecting-rod by means of a rod and rocking lever. This motion gives a quicker cut-off in the high-pressure cylinder, and a later cut-off in the low-pressure cylinder than could be obtained with an eccentric. It also allows of extra length being given to the main bearings of the engine. The working parts run in a bath of oil and water, all parts being constantly splashed over with lubricant. An inspection of the section (fig. 1329) will

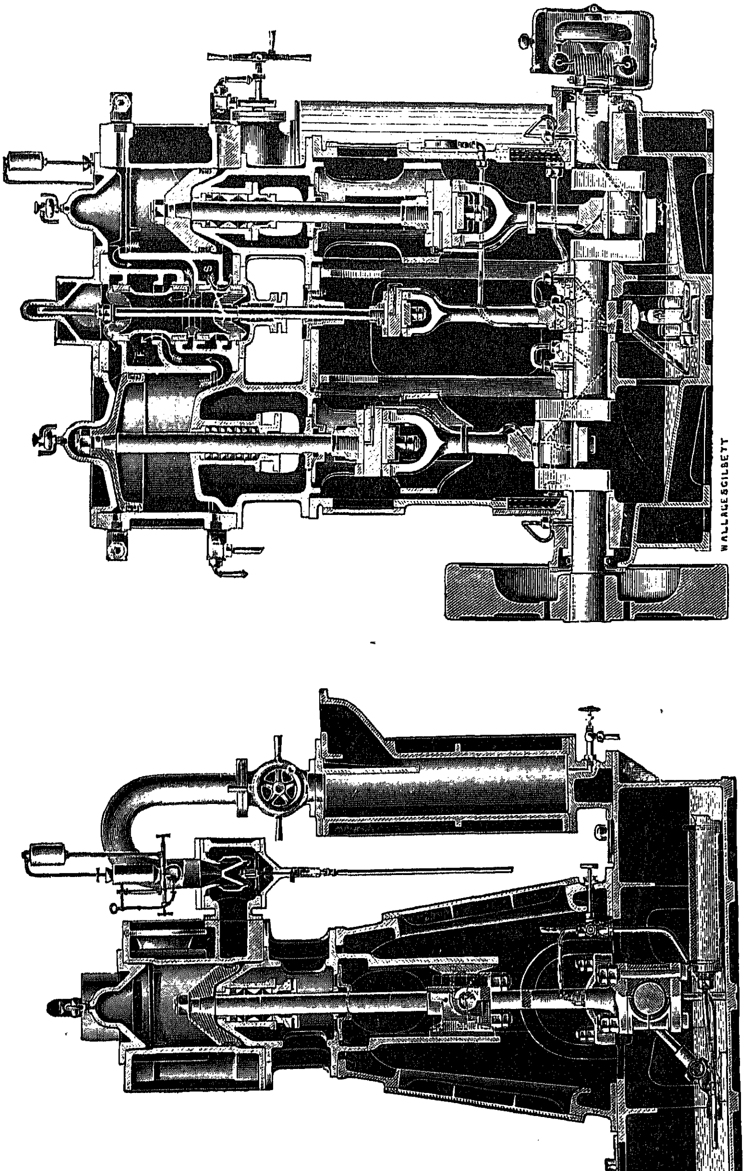


Fig. 1330.—Belliss Compound Engine

show the extreme simplicity of this engine. There are no internal glands, and of the external glands, that on the valve spindle is subject to exhaust pressure only, while that on the low-pressure cylinder is subject to a low pressure of steam only.

Engines from which fig. 1329 has been prepared are running at Folkestone electric lighting station. The following were the consumption figures obtained from the engines tested in place at Folkestone, with Parker continuous-current dynamos:—

Steam pressure	130 lbs.
Vacuum	25 inches
Revolutions per minute	300
I.H.P.	564
E.H.P.	482
Combined efficiency	85.4 per cent
Steam consumed	17.1 lbs. per I.H.P. hour
„ „	26.9 lbs. per kilowatt-hour

The Belliss Engine.—This engine is a high-speed double-acting engine, and being the first and best known of its class, is here described. The compound type, as shown by the sectional illustration (fig. 1330), has two cylinders operating cranks placed opposite to each other. The steam is controlled by a single piston-valve, or, more correctly speaking, by two piston slide-valves, the low-pressure valve being superposed above the high-pressure valve, and working in the same valve chamber. With this explanation the action is at once clear; the high-pressure valve takes steam on the inside edges, and exhausts at the upper end directly into the low-pressure valve, the exhaust from the lower end of the high-pressure slide-valve passes through the interior of the valve to the low-pressure valve. The low-pressure valve cuts off steam on the outside edges, and exhausts by the liner edges direct to a single exhaust-pipe. The triple-expansion type shown by the sectional illustration (fig. 1331) has separate cylinders and slide-valves acting on three cranks at equal angles. The slide-valves all take steam on the inside edges and exhaust on the outer edges, the exhaust steam from the upper end passing through the slide-valve to the receiver-pipe in the case of the high-pressure and intermediate-pressure cylinders, and to the exhaust-pipe for the low-pressure cylinder.

System of Lubrication.—The following description of the system of lubrication applies equally to both triple and compound engines. The oil for lubrication lies in the crank chamber, and is drawn thence by one or more pumps and delivered to the bearings through an arrangement of pipes and channels, forming a complete arterial system leading to all the bearings. From the oil pump a main oil-pipe leads along the crank-shaft with branch pipes connected to nipples on the upper half brass of each bearing. A circumferential groove is turned in each bearing, into which the oil is supplied, and from this groove an oil hole is drilled through the crank-shaft and crank-pin to a similar circumferential groove round the crank-pin brass. From this groove a pipe is carried up the connecting-rod to the cross-head pin, from which by a similar arrangement of circumferential groove oil is taken to the guide-shoe. The lubrication of the eccentric-rod is dealt with on similar lines. A pipe is led off from the main to a pressure-gauge, which shows at a glance whether a satisfactory pressure is being maintained in the system. There is also a valve for regulating the pressure. The pump is of the oscillating type without loose

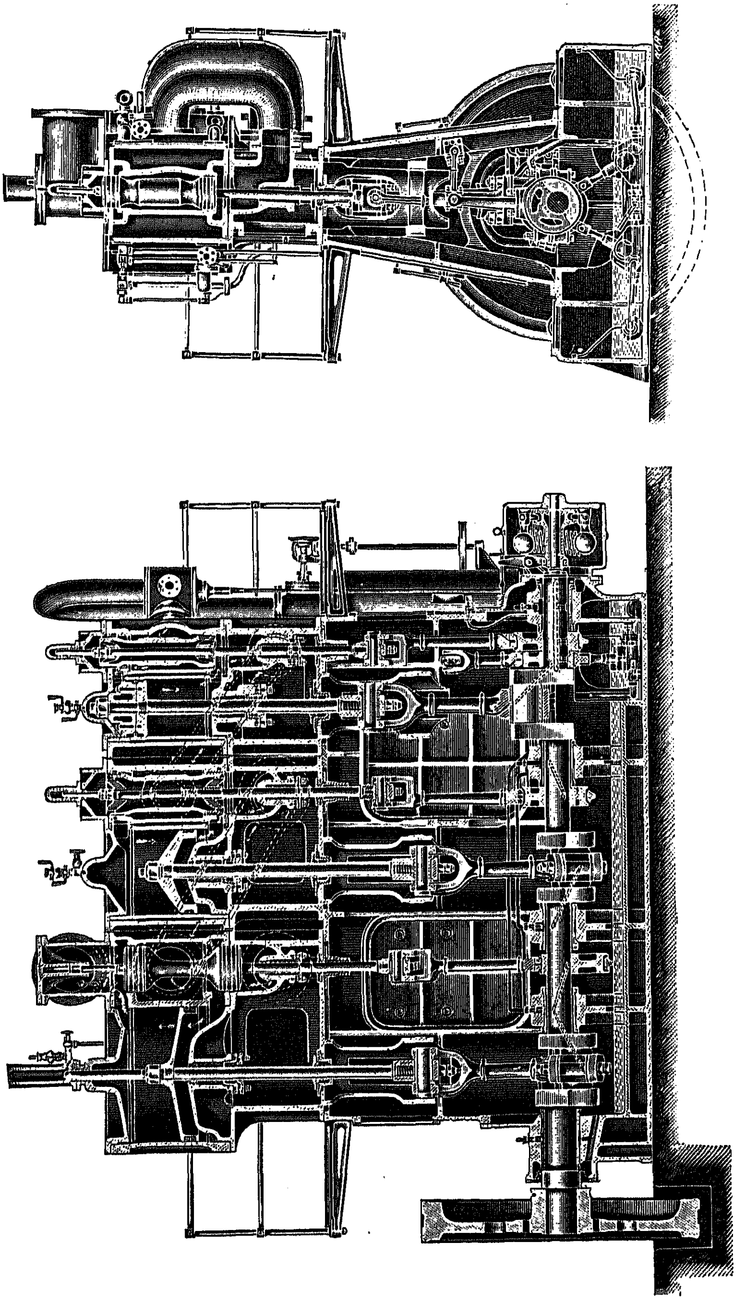


Fig. 1331.—Belliss Triple-Expansion Engine

valves or packing, the plunger being simply a working fit in the barrel, which oscillates on a trunnion. A fine wire-gauze strainer is connected to the suction-pipe of the pump. The oil used must be a pure mineral oil, not a vegetable oil, which is liable to thicken in use and cause trouble.

The pressure of the oil is usually maintained at from 10 to 20 lbs. per square inch. The invention of this very complete system of "forced" lubrication has made it possible to run double-acting engines at a high rotational speed for long periods. Before its introduction only comparatively short runs could be made before bearing brasses needed "taking up".

With regard to steam economy, the following figures were obtained from a compound condensing engine of 250 I.H.P.:—

Steam pressure at engine	150 lbs.	B.H.P.	218
" " in chest	138 "	$\frac{\text{B.H.P.}}{\text{I.H.P.}}$	94 per cent.
Vacuum	27.8 ins.		
Revolutions per minute	360	Steam consumed	18.2 per I.H.P.
I.H.P.	232	" "	19.4 per B.H.P.

With a dynamo giving an efficiency of 92 per cent, this would mean a consumption of about 28 lbs. per kilowatt-hour.

With regard to the adoption of the side-by-side class of engine in preference to the "complete engine on each crank" class, it has been seen (Chapter III) that even with unequal powers on the cranks the turning effort is still very good; the gain in economy due to employing few and large cylinders rather than many and small, is found by Messrs. Belliss & Morcom to be, roughly speaking, 10 per cent, say 1½ lb. per B.H.P. for a triple-expansion engine and 2 lbs. per B.H.P. for a compound engine.

Inverted Vertical Triple-Expansion Slow-Speed Engine.—The very complete sectional view given in fig. 1332 of an 800-I.H.P. engine built by Messrs. Mather & Platt needs no description. It is a fine example of the double-acting open vertical marine-type engine. The cylinders are carried on massive cast-iron back standards, to which planed guides are bolted, while they are supported in front by turned and polished steel columns. The valves are of the balanced-piston type. The bed-plate is of cast-iron box section, the pedestals being cast on and bored at one setting to ensure complete alignment. The governor shown is of the Pickering (throttling) type. The expansion governor shown in fig. 1288 is sometimes fitted to this class of engine. The engine illustrated is one of six supplied to the Castner-Kellner Alkali Co.'s works near Runcorn, and is used chiefly for electrolytic, but incidentally also for lighting and power purposes.

Very complete tests were carried out on this engine under ordinary working conditions at the Castner-Kellner Alkali Co.'s works, and gave the following results at a little over three-quarters load. Being a "throttle"-governed engine, the result at full load would necessarily be better still.

Boiler pressure ...	176 lbs.	Total I.H.P.	701.8
Vacuum	26.6 ins.	Kilowatt output	435
Revolutions per minute	109	Combined efficiency ...	83 per cent
I.H.P. in H.P. cylinder	228.6	Boiler feed per I.H.P. hour	14.4
I.H.P. in I.P. "	238.8	" " kilowatt-hour	23.3
I.H.P. in L.P. "	234.4		

The power being very equally divided between the three cranks, the cyclic irregularity of turning would be very small, and the engine consequently well adapted for driving alternators in parallel.

Inverted Vertical Two-Crank Compound Slow-Speed Engine.—The engine illustrated by fig. 1333 is one of two erected by Messrs. Victor Coates & Co., Limited, of Belfast, for working the trams at Newcastle-on-Tyne. It is a compound two-crank engine, having its cranks placed at 90° to each other. The valves are of the Corliss type, the cut-off being determined by a trip-gear, as is usual. The governor controls the point

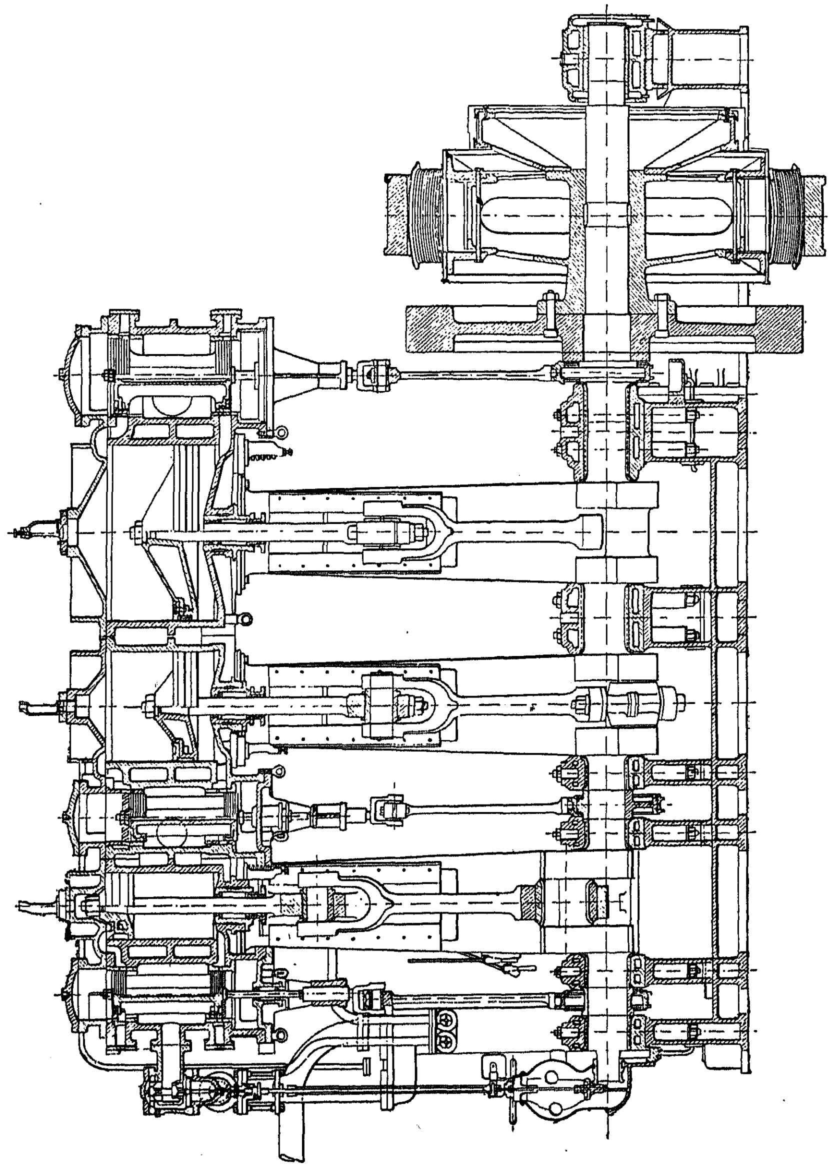


Fig. 1332.—800 I.H.P. Triple-Expansion Vertical Engine and Electrical Generator

of cut-off on the high-pressure cylinder only, that on the low-pressure cylinder being adjustable by hand while the engine is running, by the hand lever, clearly shown on the illustration. The rods and levers leading from the governor to the high-pressure trip-gear are also clearly shown. The very large fly-wheel provided for tramway work is well exemplified in this engine, the rim being 15 feet diameter, 2 feet 3 inches wide, and 1 foot 4 inches deep. This heavy wheel was found on test to be capable of controlling the *momentary variation* of speed to within 6 per cent on full load being suddenly removed, and to within 3 per cent on full load being suddenly put on. The *permanent* variation in speed from full to no load was 2 per cent. The leading dimensions of the engine are—

H.P. cylinder	22 inches diameter,	42 inches stroke
L.P. „	46 „ „	42 „ „
Dynamo and fly-wheel bearings,	16 inches diameter,	36 inches long
Engine bearings...	12½ „ „	12 „ „
Crank-pin	12½ „ „	10 „ „

The engine was designed to give 1000 I.H.P. as economical load, 1250 I.H.P. as maximum load; revolutions, 90 per minute; steam pressure, 160 lbs. in boilers; vacuum, 28 inches. The steam consumption with saturated steam, the weight of water being taken from the air-pump discharge, and including jacket steam, was—

At overload ...	15.7 per I.H.P.,	23.7 per kilowatt-hour
„ full load ...	14.7 „ „	21.8 „ „
„ two-thirds load	14.05 „ „	21.6 „ „
„ one-third load	14.8 „ „	23.4 „ „

Horizontal Cross Compound Slow-Speed Engines.—Cross compound engines usually present the same general features. They are built with or without reheaters between the cylinders, and the valve-gear and valves are either piston slide-valves, Corliss valves, or drop-valves. The drop-valve gear has been described already in connection with governing. Corliss valves and piston valves are too well known to need detailed description here. Fig. 1334 is an outline illustration of a cross compound condensing engine, built by Messrs. Pollit & Wigzell, Limited, for electric light and traction work. The general dimensions of the engine are:—H.P. cylinder diameter, 19 inches; L.P. cylinder diameter, 39 inches; stroke, 3 feet 6 inches; fly-wheel, 14 feet diameter, weight 19½ tons. At a speed of 100 revolutions per minute the engine will develop 800 I.H.P. as an economical load, and 1000 I.H.P. as an emergency load; this with a boiler pressure of 170 lbs. The tail-rod of the L.P. piston is prolonged and carries the air-pump piston of a jet condenser. The governor is a Whitehead patent enclosed governor driven by two ropes from a grooved pulley on the crank-shaft. It is claimed for this governor that, when desirable, absolute uniformity of speed at all loads can be obtained without any tendency to "hunt". For ordinary cases, however, this is not necessary, and for alternating work it is even undesirable for electrical reasons.

Vith regard to steam consumption, the following results have been fied by Mr. Michael Longridge:—H.P. cylinder diameter, $15\frac{15}{16}$ inches; cylinder diameter, $31\frac{15}{16}$ inches; stroke, 4 feet; revolutions per minute, 3; boiler pressure, 124.5 lbs. above atmosphere; vacuum, $26\frac{3}{4}$ inches; P., 393.8; steam per I.H.P. hour, 13.9 lbs.; British thermal units per P. per minute, 271.3.

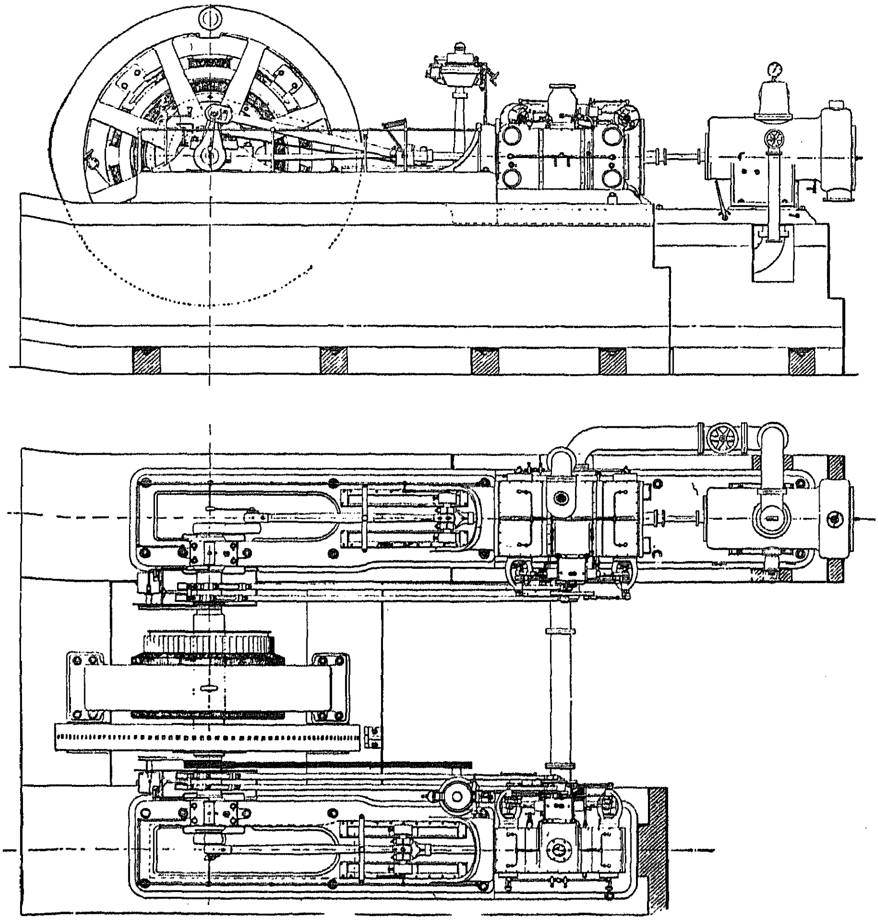


Fig. 1334.—Horizontal Corliss Engine

Fig. 1335 represents two other engines by the same makers, designed non-condensing work and for driving alternators in parallel. The nders are—H.P., 14 inches diameter; L.P., $23\frac{1}{2}$ inches diameter; stroke, nches; revolutions per minute, 100; periodicity of alternators, 80. The governors are of the Hartnell type, actuating trip-gear on the H.P. liss valves, the L.P. cut-off being adjustable by hand. The alternators h have an output of 125 kilowatts normally, and an emergency output 135 kilowatts at 100 revolutions per minute. The fly-wheel is of cast-ı, and carries the poles on its rim. The diameter over the poles is 13 feet, and the total weight of wheel and poles is about 10 tons. The *momentary* speed variation on full load being suddenly removed was 3½ per cent, and the *permanent* variation from full to no load 1 per cent. The total steam consumed with 110 lbs. pressure was 35.75 lbs. per kilowatt-hour.

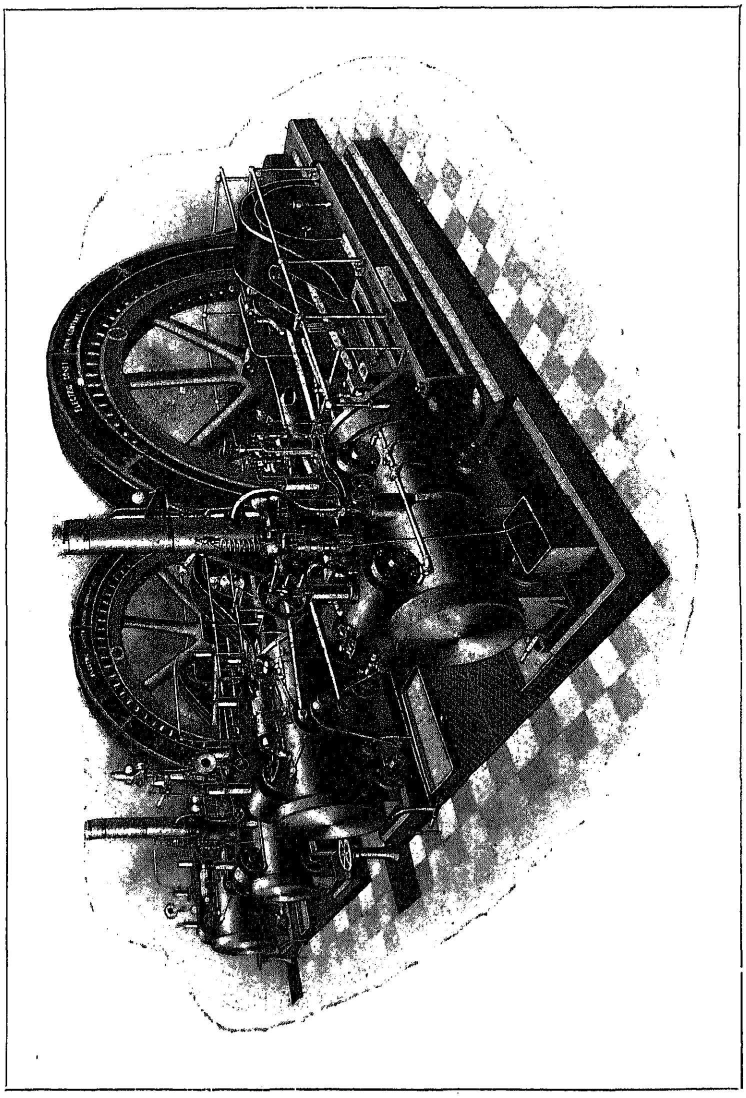


Fig. 1335.—Horizontal Corliss Engine with Alternators

A pair of vertical cross compound engines by the same makers, running with 147 lbs. pressure and 24½ inches vacuum, gave a consumption of 23.34 lbs. per kilowatt-hour, 14.4 lbs. per I.H.P.; the output of these engines being 300 kilowatts, and speed 123.5 revolutions per minute.

CHAPTER VI

STEAM-TURBINES—POWER AND EFFICIENCY

The remarks on power necessary for the proper performance of the various duties to which electric generators are put, made in connection with reciprocating engines, apply equally well to steam-turbine-driven generators. The efficiency curve, fig. 1284, does not hold for a steam-turbine, since the I.H.P. is not determinable; but the margin of output to meet the fluctuating demands for current from a tramway station is as much necessary for a turbine as for any other class of engine. The friction of a turbine should be very small indeed, since the only rubbing parts are the bearings, which are supplied with oil under pressure, the governing apparatus, and the collars on the dummy rings for balancing the end-thrust of the steam. The total friction of a reciprocating engine is made up of shaft friction, piston and piston-rod friction with cross-head friction, and valve friction. Usually the shaft-bearing friction amounts to about 45 per cent of the total engine friction. There does not seem, therefore, to be any reason why the friction of a turbine should amount to more than 50 per cent of that of an equal-powered reciprocating engine, allowing 5 per cent for friction of governing gear, oil-pumps, and so forth. On this assumption the combined efficiency of a 300-kilowatt turbo-generator would be approximately as below:—

At quarter load	81.5 per cent
„ half load	89.0 „
„ three-quarters load ...	92.5 „
„ full load	94.5 „
„ 25 per cent overload	95.0 „.

By applying these percentages to the ascertained consumption per kilowatt-hour of a turbo-generator, equivalent figures may be obtained for comparison with the performances per I.H.P. hour of reciprocating engines under similar conditions of load, steam pressure, and vacuum.

CHAPTER VII

STEAM-TURBINES—GOVERNING

There are two methods of governing turbines—

i. Governing by admitting the steam in gusts of varying duration.
ii. Governing by throttle.

Governing by Gusts.—To make the method of governing by gusts clear, the diagram fig. 1336 has been prepared. It is not a detail from any particular turbine, but is purely diagrammatic. *a* is a worm on the turbine shaft, and *b* a worm-wheel on a small shaft *d*, which carries an eccentric *c*. The wheel *b* gears with *a* and turns the shaft *d*, which makes perhaps one revolution for every ten revolutions of the turbine. The eccentric-rod *e*, by its reciprocating motion, moves the end of a lever *f* up and down. The lever *f* is pivoted at *g*, and a link *h* connects the short arm of *f* to the end of a second lever *j*. This lever *j* is in turn pivoted at *k*, and carries another link *l* at its other end. Link *l* connects on to a pin on a third lever *m*, which works about a fixed fulcrum *n*. The slide-valve *o* of the relay cylinder *p* is connected to the end of the lever *m*. As the eccentric-rod *e* moves up and down, the slide-valve *o* moves up and down, admitting steam to the relay cylinder in exactly the same way as does the slide-valve of any ordinary steam-engine. The piston *q* is connected by a rod to an admission-valve *r*, which admits steam to the turbine. So far it is evident that the admission-valve *r* will be opened and closed once every revolution of the shaft *d*, and the duration of the opening will be longer or shorter according to the setting of the slide-valve *o*. In order that the setting of the valve *o* may be automatically adjusted to the requirements of the load on the turbine, the pivot *k* of the lever *j* is under the control of a centrifugal governor *s*, which raises it or lowers it according to the position of the balls of the governor. The effect of raising or lowering the pivot *k* is to make the slide-valve *o* admit steam to the relay cylinder earlier or later, and cut off the steam later or earlier in the same way as does altering of the lap of a slide-valve. Thus it is seen that the duration of the gusts of steam to the turbine is under control of the governor. The governor itself is rotated by means of a worm and wheel on the turbine shaft or on the shaft *d*, as may be convenient. Any good form of centrifugal governor is sufficient for the purpose, and the percentage variation of speed of the

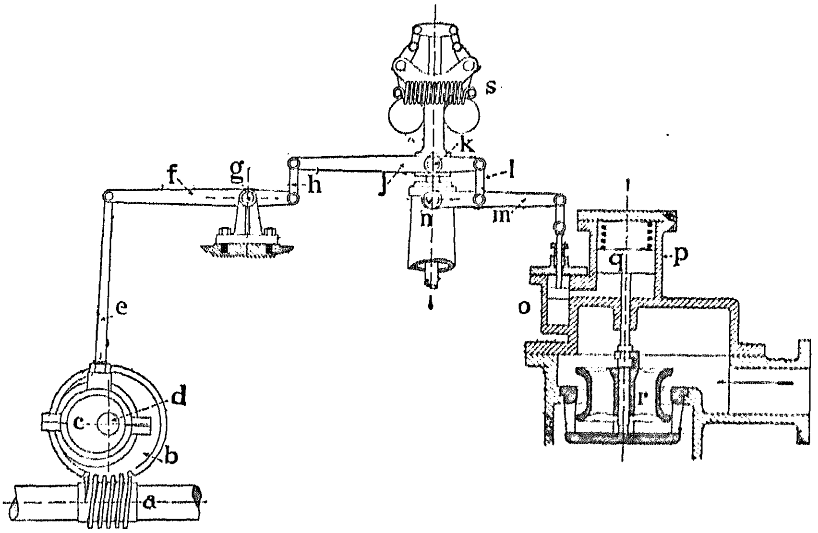


Fig. 1336.—Steam-Turbine Governor Gear

generator from full to no load may be kept within as narrow limits as with a reciprocating engine. This form is employed by Parsons.

Governing tests carried out on a Parsons turbine for the *City of Elberfeld* showed a variation of 3.6 per cent from no load to full load.

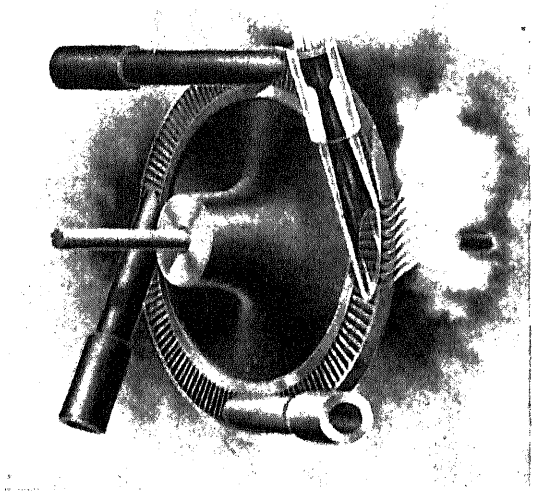


Fig. 1337.—De Laval Steam-Turbine Wheel

Governing by Throttling.—Although governing by the gust method also throttles the steam through the valve *r* at very light loads, an example of throttle governing pure and simple is to be found in the De Laval steam-turbines, where a governor on the gear-wheel shaft operates a throttle-valve in the steam admission-pipe, the turbine wheel being driven by a number of jets of steam impinging on it, as shown in fig. 1337. An extra regulation is also provided by means of an arrangement whereby the vacuum of the condenser can be varied by opening to the atmosphere, which is useful should the throttle-valve get jammed. If the speed gets above a predetermined amount a small valve is opened allowing access of air into the condenser, so reducing the vacuum produced, and consequently bringing down the speed.

A description of the De Laval turbine is given later.

CHAPTER VIII

STEAM-TURBINES—FLY-WHEELS

The enormously high speed of rotation attained by turbines, from about 5000 revolutions per minute for a 25-kilowatt set to 1500 revolutions per minute for a 1000-kilowatt machine, makes the provision of extra fly-wheel power, beyond that supplied by the turbine wheel and armature of the dynamo, unnecessary. It has been shown how, in reciprocating engines, the variation in crank effort during a revolution produces irregularity in angular velocity, and how this irregularity can be kept within bounds by the provision of sufficient fly-wheel power. In the turbine, cyclic irregularity of turning does not exist in quite the same way as in reciprocating engines. Whereas in the reciprocating engine the speed of rotation fluctuates from a maximum to a minimum a certain number of times each revolution, in a turbine governed by the admission of one gust of steam every 10 revolutions, the speed of rotation will be at a maximum and at a minimum once in every 10 revolutions. The cycle, then, with a turbine is not *one* revolution, but 8, 10, or 12 revolutions, according to the relative speeds of the turbine wheel and the governor shaft *d*, fig. 1336.

In a turbine governed by gusts, the pressure in the steam-chest just before the first ring of blades fluctuates roughly, as shown in fig. 1338.

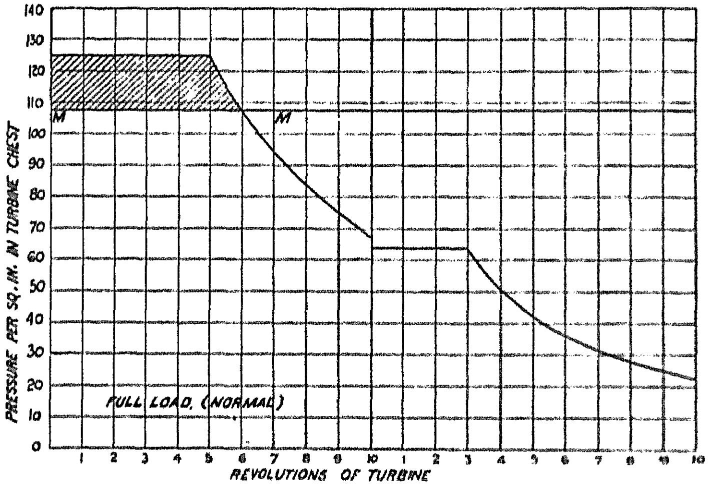


Fig. 1338.—Turning-Effort Diagram. Steam-Turbine. Coefficient of fluctuation of energy = 0.088 (as fraction of turbine energy in ten revolutions)

Since the drop in pressure between steam-chest and exhaust-pipe furnishes the energy to drive the turbine, the coefficient of fluctuation will be approximately as shown on fig. 1338, viz. $\kappa = 0.088$ expressed as a fraction of the energy exerted by the steam in 10 revolutions of the turbine.

Treating the turbine by the same method as has been adopted when dealing with the reciprocating engines, but remembering that the steam cycle is now 10 revolutions (for the case of one gust every 10 revolutions), the following figures hold:—

Case XV.

Steam-turbine.
300 kilowatts output. Periodicity of alternator, 50.
I.H.P., 426.
Revolutions per minute, 3000.

$$\theta = 2160^\circ. \quad \text{(See fig. 1340.)}$$

$$E_0 = \frac{426 \times 33000}{300} = 46860 \text{ ft.-lbs. per 10 revolutions.}$$

$$\kappa = 0.088. \quad \text{(Fig. 1340.)}$$

$$\kappa E_0 = 4123.68 \text{ ft.-lbs.}$$

$$\begin{aligned} p &= \frac{50 \times 60}{3000} = 1. \\ q &= \frac{4\,\alpha}{\theta\, p} = \frac{1}{540}. \\ E &= \frac{\kappa\, E_0}{2\, q} = 1113394 \text{ ft.-lbs.} \end{aligned}$$

W^* at 150 feet per second = 3182 lbs.
1.42 ton.

It is seen from the above that the steam-turbine is in an excellent position to ensure that a limit of 2° phase displacement shall not be exceeded.

CHAPTER IX

STEAM-TURBINES—STEAM CONSUMPTION

Theoretical Steam Consumption.—The number of thermal units due from 1 lb. of steam expanded from admission temperature to exhaust temperature is the same in all cases, whether the steam is expanded in a reciprocating engine or a steam-turbine. The formulæ for theoretical consumption given hold good, therefore, for the steam-turbine.

The pressure on the cylinder cover and piston of a reciprocating engine forces the piston away from the cover; in a turbine, pressure between two adjacent blades on the same ring could have no power to turn the turbine, since the projected area of the back of one blade is equal to the projected area of the front of the next blade, and the total pressures on the blades would balance each other were it not for the direction of flow of the steam. It is only by inclining the fixed blades in the turbine case, and curving them, so that the incoming steam is directed on to the blades of the revolving wheel in the direction it is intended the turbine shall revolve, that motion of the wheel results. The energy a pound of steam possesses in virtue of its pressure (potential energy) is converted into energy of motion (kinetic energy) as the steam passes through the turbine. In turbines, such as the Parsons, the change goes on from ring to ring of revolving blades in such a way that

$$\begin{matrix}\text{Potential energy}\\ \text{of steam}\end{matrix} + \begin{matrix}\text{Kinetic energy}\\ \text{of steam}\end{matrix} + \begin{matrix}\text{Kinetic energy}\\ \text{of rings passed}\end{matrix} = \text{Constant.}$$

The above assumes that there is no loss of energy due to the impact of the steam on the blades of the turbine. At the end of the process, when the steam leaves the last ring of blades at the pressure of the exhaust, there is only enough energy left in it to carry it into the con-

---

[*] The velocity of blades varies from about 7536 feet per minute at the small end of a turbine to about 26,400 feet per minute at the big end or low-pressure end. This is measured at the tips of the blades. Taking the velocity of the mass of the drum at half the velocity of the periphery, this means 9000 feet per minute, or 150 feet per second, as the mean velocity of the equivalent fly-wheel.

denser, the balance of potential energy having been changed into kinetic energy of the turbine wheel, and energy wasted in friction.

In turbines such as the De Laval the potential energy of the steam is entirely changed into kinetic energy before the steam reaches the turbine wheel at all; this is done in a nozzle, or nozzles, of such a shape that the steam is expanded in it from the initial pressure down to the ultimate pressure of the exhaust pipe.

Every reduction of this ultimate pressure of exhaust, or, that is to say, condenser pressure, produces a much greater increase in efficiency than is the case with a reciprocating engine. The reason for this is that in the turbine larger ranges of expansion have to be reckoned with. The steam in passing through the Parsons turbine, for instance, is expanded ten times the number of expansions in a reciprocating engine. It is not that reciprocating engines as such would not give better results at high vacua than at low, but because the low-pressure cylinder valves, passages, &c., would have to be made of such inordinate sizes to deal with the enormous volumes at low pressures, that 25 inches vacuum is generally found to be about the economical limit. It is usually the case with reciprocating plant that it is found necessary to place the condenser at some distance from the engines. In the turbine case it is usual to have the condenser directly below the turbine, the exhaust being of only a few feet in length, and can be of four to six times the area of that of a reciprocating plant of the same output; the drop in pressure between condenser and cylinder is in consequence very considerably less. In the reciprocating plant a difference in pressure of 5 inches of mercury between the absolute pressure in the condenser and the mean back pressure in the condenser is quite common, whereas in the turbine case a drop of ½ to ¾ inch may be expected. The gain in efficiency of increasing the vacuum in a condenser condensing steam from a turbine from 24 to 28 inches of mercury is about 4 per cent.

Missing Quantity.—In a turbine there can be little or no condensation, as on steady load the metal temperatures do not change. However, water, which comes over with the steam in addition to that formed by the condensation of some of the steam due to work done, produces in its passage through the turbine a large amount of friction, and thus reduces the efficiency. We can reduce this loss by superheating the steam. Dry steam will leak much less readily than wet steam, and consequently by superheating the steam at entrance it is possible to reduce steam leaks past the blade tops to a minimum, and so produce a gain in economy.

A diagram taken from the steam-chest of a turbine, between the admission-valve and first ring of blades, by an indicator whose drum is moved at a rate of speed corresponding to the velocity of the turbine wheel, would take a form somewhat like that shown in fig. 1338. In a report of tests taken on a Parsons steam-turbine, the limits of pressure recorded in the steam-chest of the turbine, just before the first ring of blades, are given in the following table.

From the figures in the table it is seen how, when the load is reduced, the governor gear reduces the range of steam pressure within the steam chest.

Load.	Pressure at Stop-Valve.	Pressure in Steam-Chest, absolute.		
		Maximum.	Minimum.	Mean.
Overload	144	126	96.2	111.1
Full load	149	125	68.2	96.5
Three-quarters load	153	108	53.4	80.7
Half load... ...	148	86	37.2	61.6
Quarter load ...	144.5	56	23.2	39.6
Excited only ...	147	Fluctuation scarcely perceptible		15.87
Friction only ...	149	No fluctuation		8.20

The pressures were measured by an indicator of which the drum was revolved by hand; and it is a matter for regret that a continuously-moving paper was not used, so that the true form of the admission curve of pressure might have been obtained. Fig. 1338 has been prepared from the data above.

Actual Steam Consumption of Turbines.—The results of a few tests of steam-turbines are here given. In order to make a comparison between the performance of the steam in the turbine and in the cylinder of a reciprocating engine, an assumption has to be made with regard to the probable mechanical efficiency of the turbine as has been explained on p. 149. It was there shown why it is reasonable to take the efficiency of a turbo-generator at full load as 94½ per cent.

Assuming that this figure holds for all turbines within the limits of power tabulated below, a fair idea is obtained as to the performance of the steam per indicated H.P.

Maker	Parsons	Parsons	Parsons
Kilowatts	19.7	100	250
Steam pressure, lbs. per sq. in.	79	130	130
Vacuum	Non-condensing	Non-condensing	Non-condensing
Revolutions per minute ...	4780	?	?
Efficiency $\frac{\text{E.H.P.}}{\text{I.H.P.}}$	(85%)	(87%)	(88½%)
Steam per I.H.P. hour, lbs....	(42.4)	(25.3)	(25.4)
Steam per kilowatt-hour, lbs.	68.5	39.0	38.0

Maker	De Laval	Parsons	Parsons	Parsons	Parsons	Parsons
Kilowatts	37.3	50	100	200	500	1000
Steam pressure, lbs. per sq. in.	114	126	127	119	140	155
Vacuum, inches	26	28	27	27	26	27½
Revolutions per minute ...	?	5044	4800	3000	2100	1461
Efficiency $\frac{\text{E.H.P.}}{\text{I.H.P.}}$	(85%)	(86%)	(87%)	(88%)	88¾%	90%
Steam per I.H.P. hour, lbs.	(20.8)	(18.0)	(17.6)	(15.8)	(15.9)	(13.6)
Steam per kilowatt hour, lbs.	32.8	28.0	26.8	24.2	24.0	20.3

The above tables are made out from data obtained from turbine trials with steam which was not superheated at all. The first series are non-condensing trials and the second are condensing, the vacuum in each case being specified.

Next let us see the variation in consumption with varying load, and for this purpose let us take the trials carried out on two Parsons turbines, the first of 500 kilowatts and the second of 1000 kilowatts.

Test of Parsons steam-turbine at Cambridge—

Capacity ... ... 500 kilowatts.
Steam pressure ... ... 150 lbs. per square inch.

In this test there was no superheating of the steam, and the turbine was driving its own air and circulating pumps.

Revolutions per minute, 2700.

Per Cent Full Load.	Steam Consumption.		Vacuum in Inches of Mercury.
	Lbs. per Hour.	Lbs. per Kilowatt Hour.	
			Barometer = 30
117	14,320	24.5	28
104	12,970	25	28
54	7,730	28.3	28
32	5,320	33.1	28
0	1,850	—	28

Test of Parsons steam-turbine at Elberfeld—

Capacity ... ... 1000 kilowatts.
Steam pressure ... ... 135 lbs. per square inch.
Revolutions per minute ... 1500.
Mean superheat ... ... 26° F.

The turbine was driving its own air-pump.

Per Cent Full Load.	Steam Consumption.		Vacuum.
	Lbs. per Hour.	Lbs. per Kilowatt Hour.	
119	23,100	19.3	27.5 inches
99	19,800	20.1	
74	16,550	22.2	Barometer, 30 inches
50	12,300	25.4	
24.6	8,310	33.7	
0	4,050	—	

The following tables give the steam consumptions of various sizes of De Laval and Parsons turbines with varying loads. In each of these cases the steam is superheated 50° F.

DE LAVAL

Size in Kilowatts.	Steam Consumption. Lbs. of Steam per Kilowatt Hour.		
	Quarter Load.	Half Load.	Full Load.
20	48	38	33
20–60	48–35	38–31	33–26
60–100	35–32	31–28	26–25
100–150	32–31	28–27	25–24
150–250	31–30	27–26	24–23

Admission pressure = 185 lbs. per square inch, absolute.

PARSONS

Size.	Average Steam Consumption in Lbs. per Kilowatt Hour.		
Kilowatts.	Quarter Load.	Half Load.	Full Load.
250	37	28	23
500	32	25	20
1000	29	22½	19
2000	27	21	17½
4000	24½	19¼	16¼

CHAPTER X

DESCRIPTION AND SECTIONS OF STEAM-TURBINES

Steam-turbines are generally divided into two classes: (1) Action or impulse turbines, and (2) reaction turbines. In the action type the steam is expanded down to its lowest pressure in the nozzles, where it attains its full velocity. It then passes into the vanes, where its velocity is reduced. The energy taken from the steam during this reduction of velocity is transformed into work on the turbine shaft. In the reaction type kinetic energy is being generated and absorbed simultaneously as the steam passes through the turbine. A much better way of classifying turbines is as follows:—

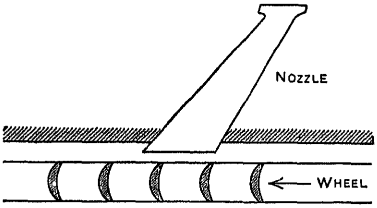


Fig. 1339 A.—De Laval Turbine Diagram

1. Turbines in which the whole energy of the steam is transformed into kinetic energy at one step. In this type the steam, after leaving the nozzle, acts on a single wheel. The wheel must in this case run at an

exceedingly high velocity, with probably large frictional losses. With this type gearing is necessary for external driving.

The representative turbine of this class is the De Laval (fig. 1339 A), to which we will return later.

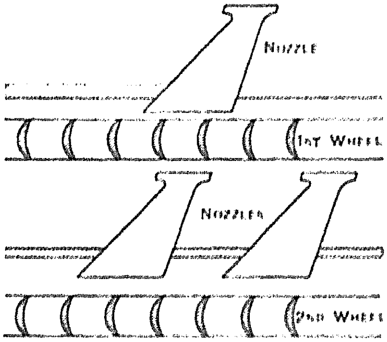


Fig. 1339 B.—Rateau Turbine Diagram

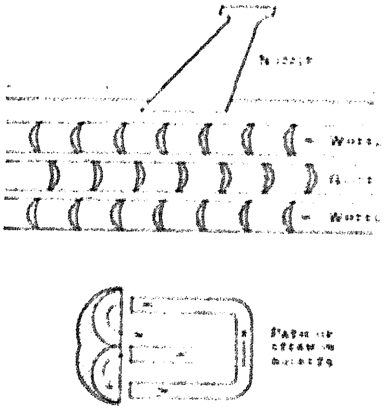


Fig. 1340 A.—Riedler-Stumpf Turbine Diagram

2. In this type the steam is expanded in several stages, with several wheels each arranged in a separate chamber, and the steam passes from one chamber to the other by means of nozzles.

The Rateau turbine is the representative of this class (fig. 1339 B).

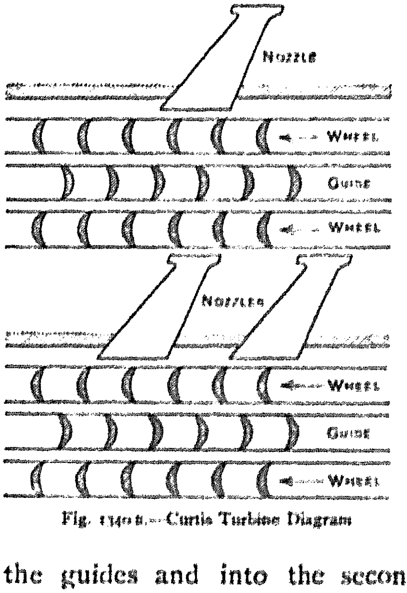


Fig. 1340 B.—Curtis Turbine Diagram

The steam velocities for the same initial and final pressures will be much less than in the preceding example, and therefore the wheels can be run much more slowly.

The Rateau turbine has from twenty to thirty stages.

3. In this type the steam expands from the initial to the final pressure on one nozzle, and then acts in succession on several sets of moving blades. Guide vanes are placed between the different moving blades to redirect the steam. The steam enters the first set of vanes with a very high velocity, the vanes are moving at a much less speed than in the type 1, and consequently only part of the kinetic energy is absorbed in the first wheel. The steam then passes through the guides and into the second moving wheel, where more energy is absorbed.

The Riedler-Stumpf is the typical turbine of this type (fig. 1340 A). In this turbine there are two sets of blades fixed side by side on the periphery of the turbine rotor drum.

Steam passes through these two sets as shown in the diagram.

4. Turbines of this type are a combination of cases 2 and 3. The steam is expanded in steps on nozzles, but between each nozzle there are at least two moving and one fixed guide vanes.

The turbine which is representative of this class is the Curtis (fig. 1340 B).

5. The fifth class of turbines has no regular nozzles. It consists of a series of alternate fixed guides and wheels. The steam expands in passing through both fixed and moving vanes, so that we can say that both act the part of nozzles. Thus the moving vanes absorb some of the kinetic energy, but are so shaped as to generate fresh kinetic energy.

The Parsons turbine is practically the only representative of this class.

It is now proposed to give a description of the Parsons turbine as used for electricity stations.

Parsons Turbine.—Fig. 1341 shows one of the compound steam-turbines made by Messrs. C. A. Parsons & Co. The revolving part of the turbine consists of a shaft upon which is mounted a drum, or drums, fitted with outwardly-projecting blades. This drum revolves in a casing fitted with corresponding but inwardly-projecting blades. The rings of blades on the drum nearly touch the case, and the blades on the case lie between those on the drum, and nearly touch the smooth cylindrical surface of it. The form of the rings of blades is shown roughly by fig. 1342. Steam entering the casing through the governor-valve finds its way into the innermost part of the case, passes through a ring of fixed guide-blades in the case, and flows in a rotational direction on to the succeeding ring of moving blades, forcing them round; it is then thrown on to the next ring of fixed blades, which again direct it in the direction of rotation on to the second ring of guide-blades; and so on through the whole turbine. The pressure of the steam diminishes gradually as it passes from ring to ring; and the drop in pressure from ring to ring supplies the energy necessary to rotate the drum. It will be noticed in the section that, in the particular turbine illustrated, the blades are in seven groups, each group larger in diameter and with deeper blades than the preceding group. The pressure of the steam on the guide-blades would force the shaft bodily to the right, but for the fact that at the left-hand end of the turbine there are a series of grooved pistons or dummies, upon which the pressure of the steam acts, and so balances the end-thrust on the blades. The connection for steam between the rotating blades and the dummies is clearly shown on the section. These dummies also act as steam-tight joints, as they are grooved as well as the case in which they turn, and the clearance between the grooves can be adjusted longitudinally. The bearings are of the tubular pattern, and are supplied with oil under pressure. The armature of the generator is coupled direct to the turbine spindle by means of a steel sleeve or flexible clutch coupling.

Governing is effected by admitting to the turbine a series of gusts of steam by the periodic opening and closing of a double-beat valve, which is moved by a steam relay connected to the turbine shaft. The duration of each gust is controlled either by an electric governor or by a centrifugal governor driven by a worm on the turbine shaft. The electrical governor, by means of a solenoid connected as a shunt to the field-

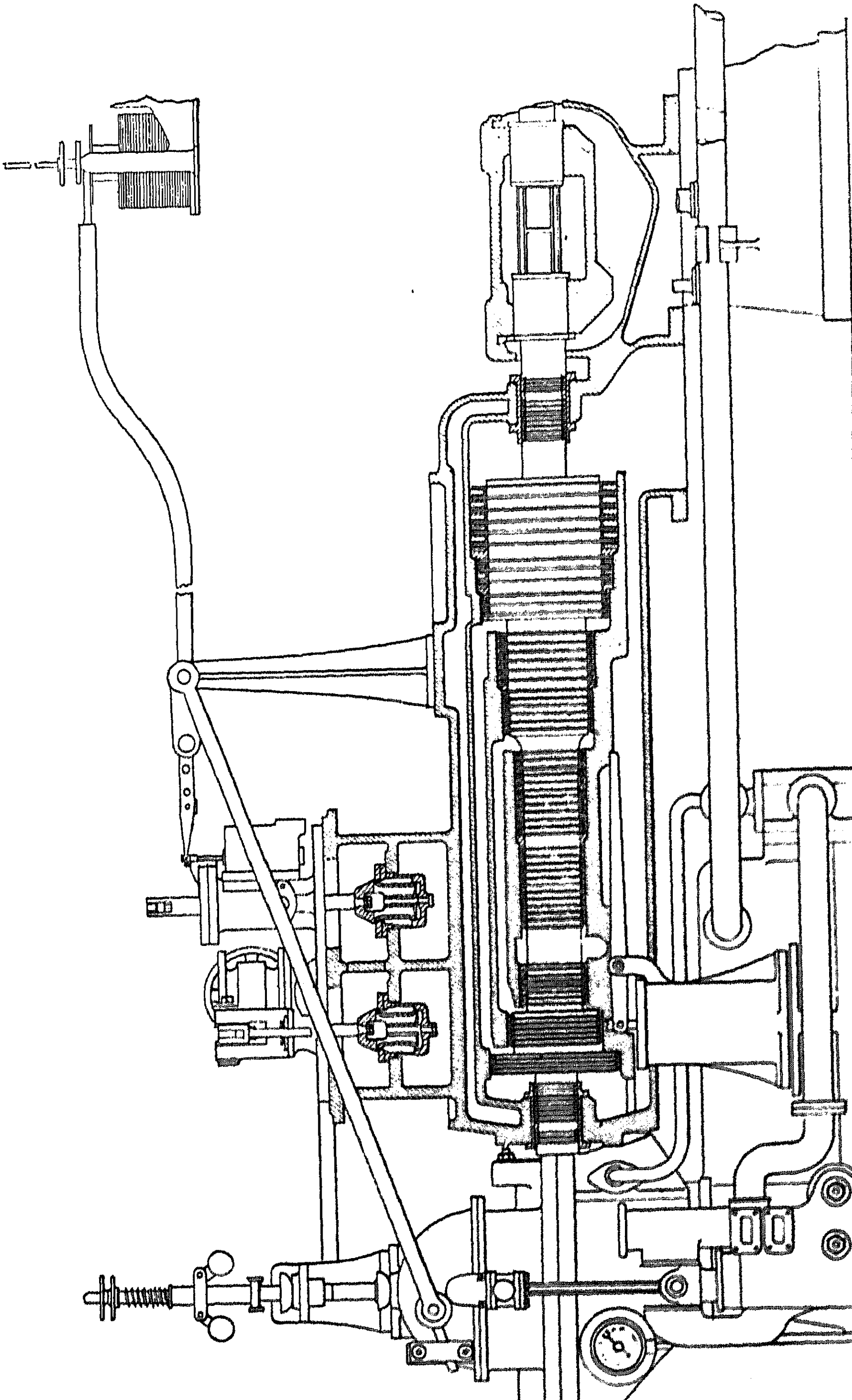


Fig. 141.—Section of the Parsons Compound Steam-Turbine as applied to Driving Dynamos

magnets, raises and lowers the end of a long lever. The centrifugal governor performs the same operation. This lever carries a fulcrum, which is thus raised or lowered according to the position of the solenoid core or of the governor balls. By the raising or lowering of this fulcrum the relay cylinder obtains steam for a greater or less time, so that the admission-valve is opened for a longer or shorter period.

Although at first sight this governing gear may seem to be of the cut-off type, an examination of curves showing total steam consumption show that they follow the Willans straight-line law, which is characteristic of throttle-governed engines. A second governor exactly like the first controls a second double-beat emergency valve, which only comes into play if perchance the main governor fails to act. The emergency valve can also be controlled by hand.

In order to take advantage of the increase of efficiency of a turbine using a high vacuum, the Parsons steam-turbine makers have introduced what they call a vacuum augmentor. In endeavouring to create a good vacuum, the first point is to avoid all air leaks, and this is easily accomplished in a turbine plant, as there are no packed glands and stuffing-boxes to leak. The only places where leakage of air is possible are where the turbine spindle comes out of the cylinder, and here leakage of air is rendered very small by packing the glands with steam, so that any leakage which takes place is steam and not air. With regard to extracting the air a great improvement has been effected by the use of a vacuum augmentor which has been recently introduced. By means of this improvement a vacuum of 27½ to 28 inches can be readily obtained. This apparatus is described in the Condenser section.

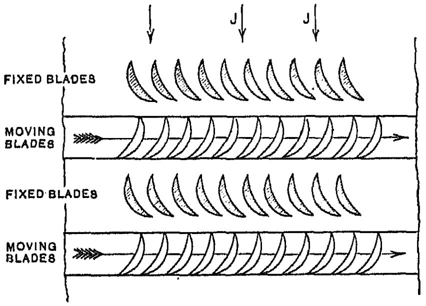


Fig. 1342

De Laval Turbine.—Fig. 1339A shows in diagrammatic form the arrangement of nozzles and the wheel of a De Laval turbine. The principle of this turbine consists in using the kinetic energy of the steam immediately after it has expanded. To accomplish this the high-pressure steam, before coming in contact with the wheel, passes through one or more nozzles which are placed at an angle to the vanes of the turbine wheel. The internal form of these nozzles has the effect of expanding the steam from boiler pressure to the ultimate pressure of the exhaust-pipe, and consequently of increasing its speed.

This turbine is governed by throttling the steam by means of an ordinary throttle-valve in the steam-admission pipe; and it is claimed that, by closing one or more of the nozzles of the turbine to suit the variations of load, less throttling is necessary than if all nozzles were left open, and the admission throttle-valve relied upon to do the governing. The advantage is that by this method the economy of steam is maintained at low loads.

The velocity of the steam issuing from the nozzles is enormous, and consequently the turbine wheel has to run at a very high speed, from 30,000 to 10,000 revolutions per minute, according to the size of the wheel and the power developed. The turbine wheel is mounted on a spindle which has two very accurately cut pinions upon it; the teeth of these pinions slope right- and left-handedly for balancing purposes. The pinions gear with two wheels mounted on a second shaft which drives the dynamo. The speed of the second shaft is from 3000 revolutions per minute for a 5-brake-H.P. turbine, to 750 revolutions per minute for a 300-brake-H.P. turbine. More often two dynamos are driven by one turbine, as shown in fig. 1343, which shows this class of engine with the series of nozzles which may be opened or closed to suit the load.

CHAPTER XI

GAS-ENGINES—POWER AND EFFICIENCY

Generally speaking, the gas-engine, which is the only prime mover other than steam-engines which is used for generating electricity on a large scale, is not mechanically so efficient as the steam-engine. That this is so is a matter of common knowledge, and the reason why is not difficult to understand. Modern steam-engines do not work, as a rule, with more than 200 lbs. initial pressure on the piston, and more often with from 120 to 160 lbs. per square inch. It is an exception for a gas-engine to work with less than 180 lbs. per square inch initial pressure, and more often they carry from 200 to 300 lbs. per square inch. For this reason the working parts have to be made stronger and bigger in a gas-engine than in a steam-engine giving the same power per stroke. Further, the greater number of gas-engines work with what is called the Otto cycle, which secures only one power-giving stroke in the cylinder every two revolutions, so that the piston is dragged four times the distance for the same power development that it would be in a steam-engine (with the same length of stroke), in which each stroke is a power-stroke.

Some gas-engines are built now, and doubtless many more will be built, for electric purposes, in which every stroke is a power-stroke, *i.e.* they are double-acting engines. These engines must necessarily be in a better position as regards mechanical efficiency than the Otto-cycle engines, for although they have to drag back and forwards their air and gas compressing pistons (the duties of which are performed by the main piston in Otto engines), these pistons with their rods are comparatively light, and can be worked by smaller pins and shafts, since they do not have to withstand the heavy initial pressures which the power pistons have to transmit. The double-acting gas-engine, then, should take up a position intermediate between the Otto-cycle gas-engine and the steam-engine as regards mechanical efficiency.

Otto-cycle gas-engines give a mechanical efficiency of between 80

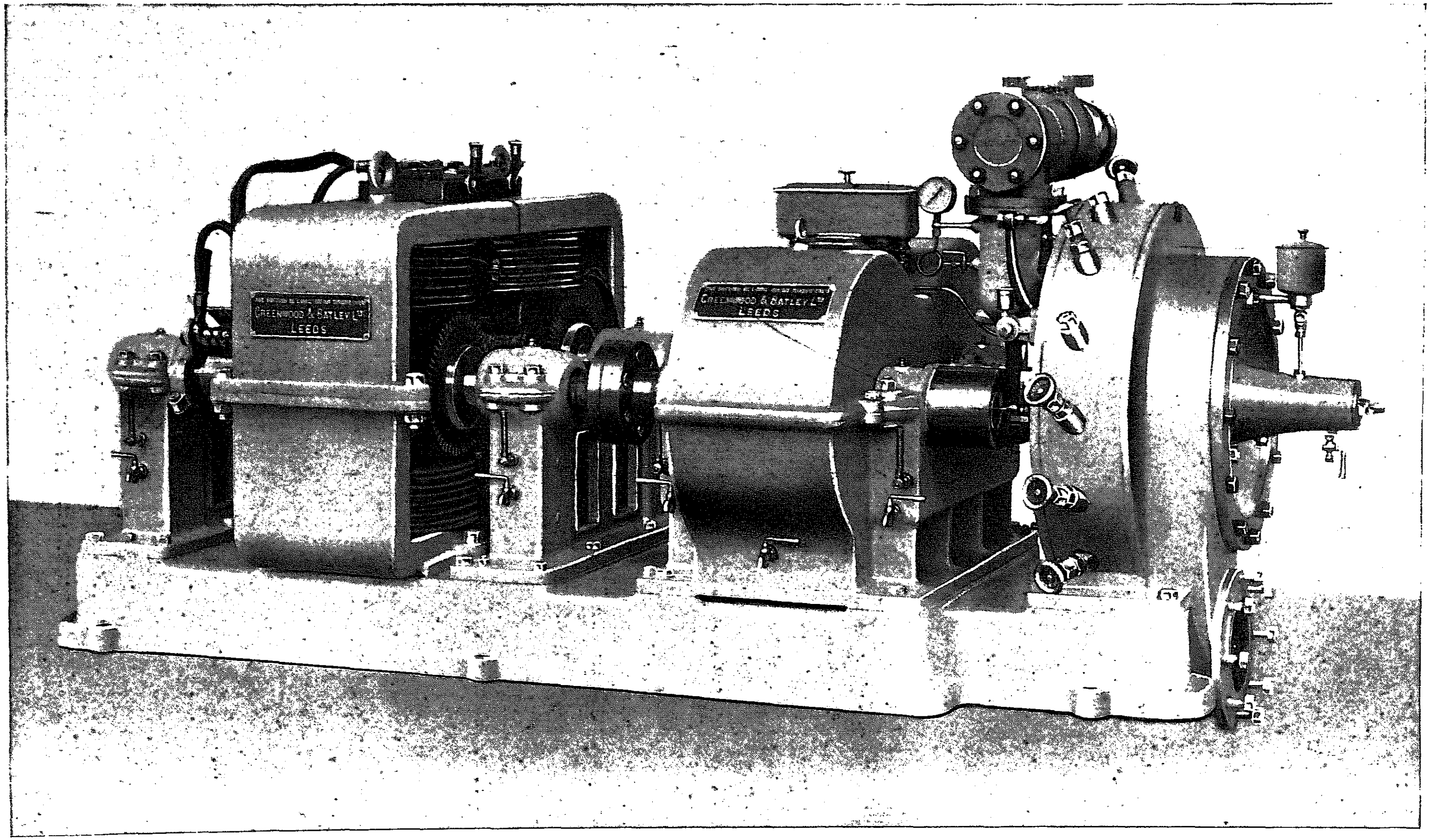


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Fig. 1343.—DE LAVAL STEAM TURBINE WITH TWO GENERATORS

and 85 per cent. The remarks that were made in connection with the power demanded from steam-driven plants hold equally well for gas-driven generators, but whereas the combined efficiency of a 300-kilowatt steam dynamo may well be expected to be as high as 83.6 per cent, $\frac{\text{E.H.P.}}{\text{I.H.P.}}$, a 300-kilowatt gas-driven generator would more probably reach 80 per cent combined efficiency, and possibly not more than 78 per cent.

CHAPTER XII

GAS-ENGINES—GOVERNING AND IGNITION METHODS

Governing Method.—In the early days of gas-engines governing was effected on the "hit-and-miss" principle, that is, the governor would allow the gas valve to be opened once every two revolutions only when the engine was running at full power. If less power was demanded of the engine the speed of the engine would increase, and the governor prevented the opening of the gas valve until the speed had again sunk to the original figure. This was a very satisfactory method of governing from the point of view of economy of gas, but left much to be desired in the direction of steady running. The engines were "hunting" in point of fact at anything below maximum load. This "hunting" is, of course, quite inadmissible for driving electric generators, and "hit-and-miss" governing has almost disappeared.

There are two methods of governing by which an explosion of the gas in the cylinder may still be obtained every second revolution: they are: (1) by varying the *quality* of the mixture of gas and air admitted to the cylinder; (2) by varying the *quantity* of the mixture admitted. By the first method, that of varying the quality of the mixture, a certain amount of variation of load may be dealt with, but it does not do for wide limits, as a weak mixture may fail to explode at all, and so the disadvantages of the "hit-and-miss" governor are introduced at quite light loads. The second method, that of varying the quantity, is the one most generally adopted for electric purposes, for although it has disadvantages in the way of economy it is *certain in its action.*

Messrs. Crossley Bros. have adopted a governor-gear whereby the quality of the mixture remains the same at all loads, but the quantity is automatically controlled by the governor. A cut-off valve worked by an eccentric from the side-shaft is placed between the air and gas suction-valves and the admission-valve. The admission-valve is opened and closed in the ordinary way, and at the same points of the stroke at all loads; the setting of the cut-off valve is, however, altered by the governor. The governor determines the position of the fulcrum of the lever to which the valve-spindle is attached much in the same way as the governor of a steam-turbine determines the position of the fulcrum of the relay cylinder slide-valve (see fig. 1338). By this means the inlet of gas and air is stopped earlier in the suction stroke at light loads, and the quantity of the mixture is reduced. The details of construction are, of course, different from those shown in fig. 1338 for turbine governing, but the effect on the "cut-off" of altering the position of the fulcrum of the cut-off valve lever is the same in the two cases.

The Westinghouse Company also regulate by varying the quantity of mixture admitted to the cylinder; this they do by throttling the admission during the suction stroke. Fig. 1344 shows the governor and connections on their three-cylinder inverted vertical engines. The governor is mounted on the vertical spindle which drives the admission-cams, and its motion is transmitted to the regulating valve through a rock shaft and levers. The gas enters through the pipe and valve shown into the mixing valve chamber through one or two openings placed on either side of the gas-supply pipe, a little below and at right angles to it. At the top and bottom of the mixing valve chamber are horizontally moving levers with pointers swinging over graduated arcs. The upper lever controls the gas supply and the lower one the air supply. The ratio between the readings on the two scales with the levers in any fixed position shows exactly the proportions of air and gas in the mixture supplied to the cylinders. The governor is of the ordinary centrifugal type. As the balls move in or out, with changes of load, the regulating valve moves up or down, increasing or decreasing the quantity of the mixture without altering its quality.

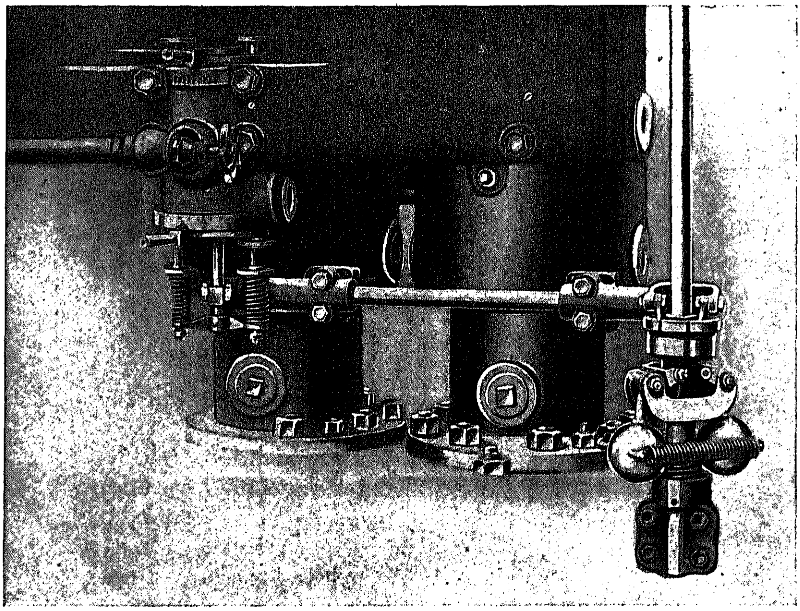

Fig. 1344.—Westinghouse Gas-Engine Governor

In the Körting double-acting gas-engine there are two pumps by the side of the power cylinder, one to supply air and the other gas. The air-pump sends a blast of pure air through the power cylinder immediately the exhaust opens, so as to clear out the burned gases; the gas-pump, which is on the same rod as the air-pump, is in the meantime driving its charge of gas back into the gas main (or by-passing it to the other side of its piston) until the closing of a valve prevents this return of gas to the main, and the rest is forced into the power cylinder with the air from the air-pump. By special construction of the power cylinder, the incoming gas is given a whirling motion so as to keep it near the admission-valve, and prevent it mixing too much with the scavenging charge of air which preceded the admission of the explosive mixture. By this means it is claimed that the charge near the admission-valve is maintained approximately of a constant *quality* at all loads.

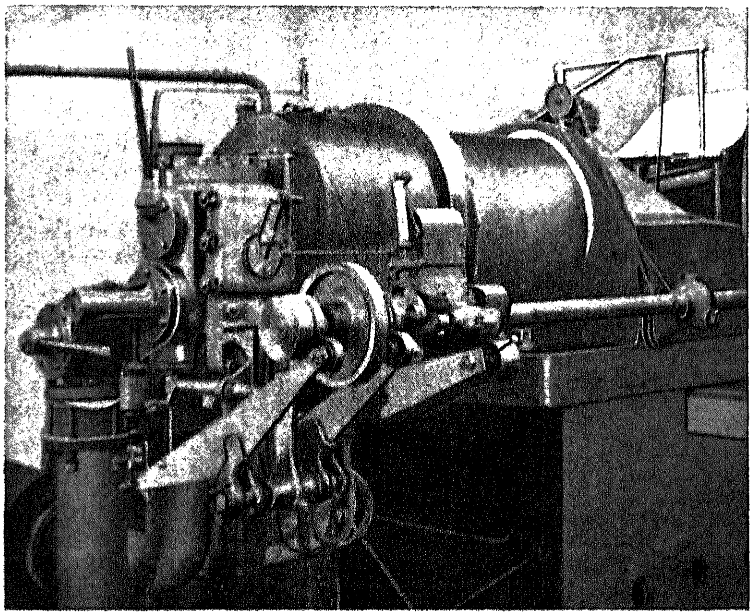

Fig. 1345.—Magnetoelectric Igniter

Ignition Methods.—"Flame" ignition, by means of a slide-valve with a pocket in it full of burning gas, opening to the cylinder at the proper moment, has been superseded by "hot-tube" ignition and by "electric" ignition. In large gas-engines using hot-tube ignition it is usual to provide a mechanically operated timing valve, which permits a flow of compressed gas and air from the power cylinder into the hot tube where it is ignited, and the flame extends back into the cylinder and ignites the charge there. These timing valves are generally opened by being pushed by a lever actuated by a cam on the way-shaft, and they are closed by springs. Occasionally these valves stick open, and when this happens there is pre-ignition of the charge in the cylinder, with a resulting impulse in opposition to the direction of rotation of the engine. The tubes also require renewal rather frequently. The electric method of ignition is now much more frequently employed, especially in large gas-engines. It has several advantages, amongst which may be mentioned:—Safety, and the saving of the outside flame necessary to keep tubes hot; less liability to pre-ignitions of the charge; more certain ignition of the charge, since the spark will ignite weaker mixtures than the hot tube; less liability to get out of order.

Messrs. Andrew & Co., Limited, makers of the Stockport gas-engine, adopt the magnetoelectric igniter. Fig. 1345 gives a view of their compact little generator, which consists of powerful horse-shoe permanent magnets, a stationary armature, and a soft-iron envelope, the last-mentioned having an alternating motion and working between the magnets and the armature. As the cam-shaft revolves, a cam raises a lever, which by means of a rod breaks contact at the moment when the spark is required. This contact-breaker is constructed as follows:—A cast-iron plug is fitted to the combustion-chamber. In this plug there are two spindles, one fixed and insulated and connected directly by an insulated wire to the magneto machine, the other constructed so as to revolve. This latter spindle has at both ends a crank, and a light spring on the outside holds it in such a position that the crank on its inner end is always in contact with the fixed spindle until the break occurs. At the moment when the tension in the alternating magnets is greatest the cam suddenly releases the magnet lever, and the rod strikes the crank and breaks contact. Provision is made for altering the time of ignition when starting the engine. The power required for working the generator is inappreciable, the mechanism simple, and there is little to get out of order; no attention is needed, and a spark of equal intensity is obtained at all speeds. An important advantage is that the electric current is only produced at the time of ignition. This ensures that there can be no ignition due to the igniter except at the predetermined period.

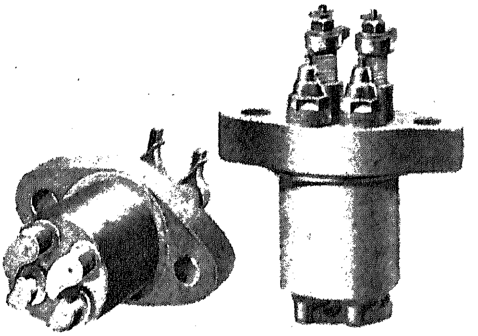

Fig. 1346.—Westinghouse Electric Igniter

The ignition of the charge in the Westinghouse engine is also accomplished by electrical means. Fig. 1346 clearly shows the igniter. The igniter has two binding-posts for the wires and two sets of current-breakers, one set only being used at a time. In case of accident or failure of the set in use, the other is in reserve and ready for immediate service. The wire supplying current can be connected to the other binding-post, and the engine run until it is convenient to shut down long enough to remove the igniter for inspection or repair. The current for the igniter is furnished by a battery through an induction-coil. The battery is connected up through a three-point switch, so that the direction of the current through the igniting apparatus may be reversed at will. Whenever the terminals on the igniter break the electric circuit the passage of the spark carries off infinitesimal particles of the substance of one terminal and deposits them on the other, the action following the direction of the current. By periodically reversing the direction of the current, whatever material has been taken away from one terminal and deposited on the other is restored, and consequently the life of the igniter is lengthened. The battery switch is fitted with a locking device, such that when the battery is cut out when the engine is shut down, it can only be cut in again in the way that will send the current in the reverse direction from what it was on the previous run.

CHAPTER XIII

GAS-ENGINES—FLY-WHEELS

With the object of showing the position of the gas-engine in respect of regularity of turning, the following diagrams have been prepared, and the coefficients of fluctuation of energy for various types tabulated. It will be remembered that for reciprocating steam-engines the coefficient of fluctuation of energy was expressed as a fraction of the energy developed per revolution, so as to include single-crank single-acting engines. For steam-turbines the cycle was taken as 10 revolutions for the case where one gust of steam is admitted to the turbine every 10 revolutions. For gas-engines the coefficient of fluctuation of energy is given as a fraction of the energy developed by the engine in *two* revolutions, to include single-cylinder Otto-cycle engines. The method followed is exactly the same as described in Chapter III, and need not be again described here. Fig. 1347 is the indicator diagram assumed to have been obtained from the power cylinder of each engine considered. The exhaust and suction strokes are taken as at atmospheric pressure, partly because it would not differ very considerably from atmosphere, and also because the diagram may then be assumed to have been taken from a double-acting gas-engine of the Körting type without any great error.

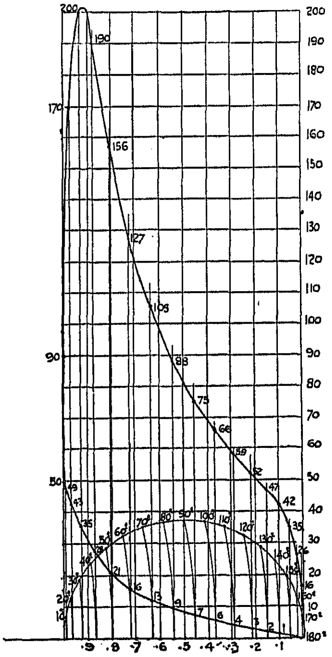

Fig. 1347

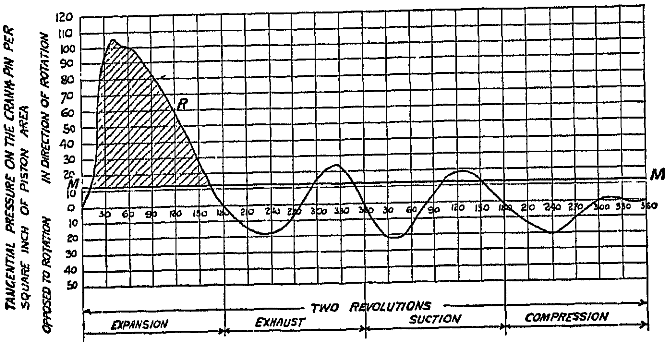

Fig. 1348.—Crank-Effort Diagram. Single-Cylinder Otto Cycle Gas-Engine. Coefficient of fluctuation = 0.920

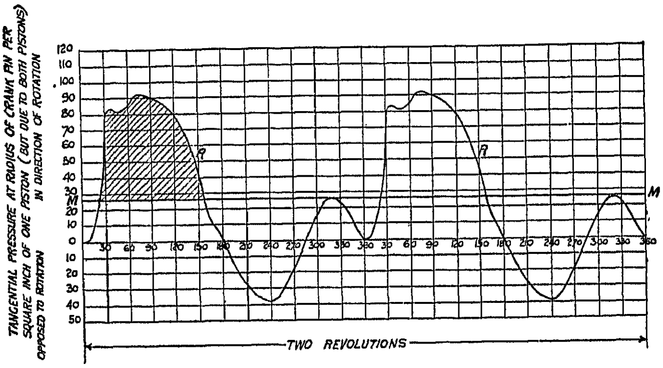

Fig. 1349.—Crank-Effort Diagram. Twin-Cylinder Otto Cycle Gas-Engine. Coefficient of fluctuation = 0.352

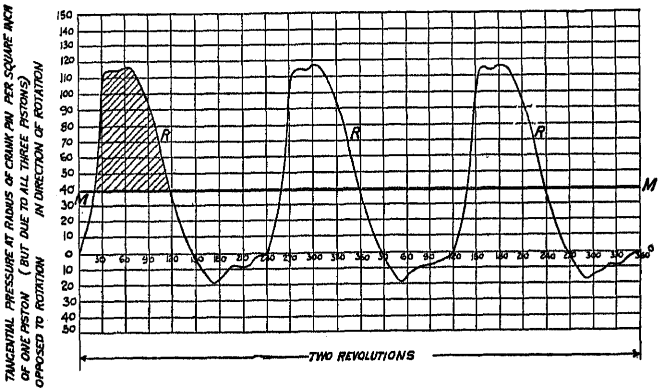

Fig. 1350.—Crank-Effort Diagram. Three-Cylinder Otto Cycle Gas-Engine—Three Cranks at 120°. Coefficient of fluctuation = 0.192

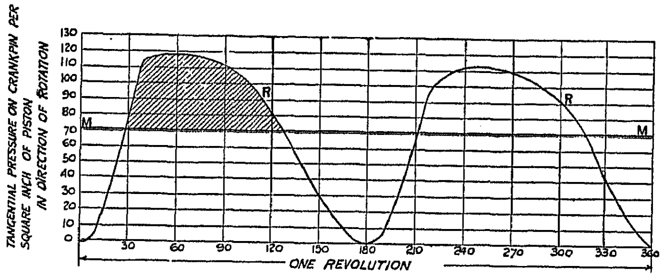

Fig. 1351.—Crank-Effort Diagram. Double-Acting Single-Cylinder Gas-Engine. Coefficient of fluctuation = 0.07 (as factor of engine energy in two revolutions)

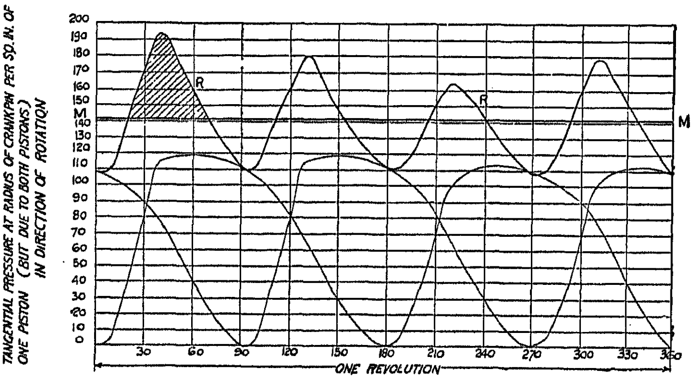

Fig. 1352.—Crank-Effort Diagram. Double-Acting Twin-Cylinder Gas-Engine—Two Cranks at 90°. Coefficient of fluctuation of energy = 0.014 (as factor of engine energy in two revolutions).

	Coefficient of Fluctuation of Energy (as fraction of energy in two revolutions).
Case XVI.—Single-cylinder Otto engine, 1 impulse every 2 revolutions (fig. 1348), $\theta = 155°$ ...	0.920
Case XVII.—Twin-cylinder Otto engine, 1 impulse every revolution (fig. 1349), $\theta = 135°$	0.352
Case XVIII.—Three-cylinder Otto engine, cranks at 120°, 3 impulses every 2 revolutions (fig. 1350), $\theta = 110°$	0.192
Case XIX.—Single-cylinder double-acting engine, 2 impulses every revolution (fig. 1351), $\theta = 100°$	0.070
Case XX.—Twin-cylinder double-acting engine, two cranks at 90°, 4 impulses every revolution (fig. 1352, $\theta = 50°$	0.014

The following table gives the equivalent fly-wheels necessary to ensure that a 2° phase displacement between two alternators in parallel shall not be exceeded:—

TABLE V

Case	XVI	XVII	XVIII	XIX	XX
Phase displacement ϕ	2°	2°	2°	2°	2°
Angular displacement for bipolar alternator α	1°	1°	1°	1°	1°
Periodicity per second	50	50	50	50	50
Output, kilowatts	300	300	300	300	300
I.H.P.	515	515	515	503	503
Revolutions per minute	100	150	150	100	120
Number of poles	60	40	40	60	50
Duration of biggest impulse ...	155°	135°	110°	100°	50°
Coefficient of fluctuation of energy κ	0.920	0.352	0.192	0.07	0.014
Reference to diagram number ...	1348	1349	1350	1351	1352
Engine energy in two revolutions, E_0 ft.-lbs.	339,900	226,600	226,600	331,980	276,650
Limit value of fluctuation of speed q	$\frac{1}{1162}$	$\frac{1}{675}$	$\frac{1}{318}$	$\frac{1}{750}$	$\frac{1}{344}$
Necessary energy in equivalent fly-wheel, E ft.-lbs.	181,683,348	26,920,012	6,916,800	8,714,250	666,588
Weight of equivalent fly-wheel at 80 feet per second, tons ...	859	127	32.7	41.2	3.1

From the above it is evident that a single-cylinder Otto-cycle engine is incapable of being provided with sufficiently large fly-wheel to ensure a phase displacement 2°, but that Cases XVIII, XIX, XX can readily be arranged to do so.

CHAPTER XIV

GAS-ENGINES—ECONOMY

The consumption of a gas-engine is very often given as so many cubic feet of gas used per I.H.P. per hour. At the time when gas-engines were only built for small powers, and the gas used was taken from the ordinary town gas mains, no doubt this gave all the information necessary to make comparisons as between gas-engine and gas-engine; it was on a par with speaking of the steam consumption of a steam-engine as so many pounds of steam per I.H.P. per hour. Both expressions are unscientific, because the heat value of a cubic foot of gas varies in different towns, and also from time to time in the same town. The heat value of a pound of steam also varies according to the circumstances of its production. The best way of giving a true measure of performance of a gas-engine or a steam-engine is to express it in terms of the thermal units supplied per I.H.P. per hour, or better still, per B.H.P. per hour. Another good way of comparing performances is to give the thermal efficiencies of the engines.

Until the production of gas for gas-engine purposes was taken in hand and brought to its present pitch of perfection, the gas-engine stood no chance as a prime mover against any good ordinary steam-engine; not because the gas-engine itself was uneconomical, but because the cost of a thermal unit supplied to it was greater than the cost of a thermal unit supplied to a steam-engine. The gas-engine, as a piece of mechanism also, was inferior to the steam-engine. Latterly, the inventions of Mr. Dowson and Dr. Mond have led to the cheap production of gas for power purposes, and the gas-engines themselves have been greatly improved as machines, so that electric generators are now being driven by gas-engines for the production of electricity on a large scale.

Theoretical Consumption.—Roughly speaking, the values of the total heat energy given out by the complete combustion of—

1	cubic foot of town gas is	680	British thermal units
1	„ Dowson gas is	160	„ „
1	„ Mond gas is	145	„ „

The consumption per I.H.P. per hour of a gas-engine using town gas is obtained from the formula—

$$v = \frac{2545}{680} = 3.74 \text{ cubic feet,}$$

assuming that the whole heat energy of the gas can be utilized as work on the piston.

A test of a Crossley gas-engine of 20 I.H.P. gave a consumption of 21.2 cubic feet of town gas per I.H.P. hour, which showed that only 17.7 per cent of the total heat energy of the gas was obtained as work in the cylinders. More recent engines by the same makers convert 25 per cent of the heat energy in the gas into work. Again, a test of a 600-horse-power engine using Mond gas gave a consumption of 52.1 cubic feet per I.H.P. per hour. The heat value of the gas being about 145 British thermal units per cubic foot, the theoretical consumption would be 17.55 cubic feet per I.H.P. per hour; hence the absolute efficiency of this engine was 33.7 per cent.

The point that has to be determined is, whether such performances are as good as can be obtained or not; and if not, how far short they are of the theoretically possible. Those who are interested in the matter cannot do better than study Mr. Dugald Clerk's work on the *Gas and Oil Engine*; all that it is possible to do here is to indicate roughly why, like steam-engines, gas-engine performances do not attain to the theoretically possible. In the first place, then, Is it possible to realize all the heat energy given to the engine as work on the piston? The well-known formula for the efficiency of a heat-engine is—

$$E = \frac{H - H_1}{H},$$

where H is the heat given to the engine,
H_1 is the heat rejected by the engine.

To obtain $E = 1$, therefore, the heat rejected (H_1) must be zero. Everybody knows that gas-engines must, for practical reasons, have their cylinders kept cool by a circulation of cold water round them. This abstracts heat from the metal, and therefore H_1, for this amongst other reasons, cannot be zero.

Taking the case of the Otto-cycle engine alone, with expansion of the gas down to the volume originally taken in, the heat supplied to the engine is—

$$H = K_v (T - t_c);$$

the heat rejected by the engine in the exhaust gas is—

$$H_1 = K_v (T' - t),$$

where K_p is specific heat of the mixture at constant pressure,
K_v is specific heat of the mixture at constant volume,
T is the temperature of the explosion,
t_c is the temperature reached by the compression of the gas and air in the cylinder before explosion,
T′ is the temperature reached by the gas after expansion,
t is the temperature of the air taken in.

The efficiency of the cycle is therefore—

$$E = \frac{K_v (T - t_c) - K_v (T' - t)}{K_v (T - t_c)} = 1 - \frac{T' - t}{T - t_c}$$

For adiabatic expansion and compression—

$$\frac{T'}{T} = \frac{t}{t_c},$$

so that—

$$\frac{T' - t}{T - t_c} = \frac{T'}{T} = \frac{t}{t_c};$$

therefore—

$$E = 1 - \frac{t}{t_c}.$$

If v_c is the volume of the compression space,
and v_0 is the volume of the compression space plus piston displacement,

and $\gamma = \frac{K_p}{K_v} = 1.4$ approximately,

$$E = 1 - \left(\frac{v_c}{v_0}\right)^{\gamma - 1}$$

From this it is seen that the theoretical efficiency of an Otto-cycle engine depends upon the amount of compression of the mixture before explosion takes place, and consequently upon the amount of clearance in the cylinder.

As in the steam-engine, so in the gas-engine, theoretically the greatest thermal efficiency is to be obtained by expanding the products of combustion down to the temperature of the back pressure. This, however, is practically inadvisable, since the mean pressure on the piston would become small, and the engine correspondingly big and costly for its output. Further, in small engines at all events, the loss of heat to the cylinder walls would, after a certain stage of expansion had been reached, exceed the gain due to the increased expansion. In the case of the Otto engine previously mentioned, compressing up to 48 lbs. absolute pressure before ignition, the ratio of clearance volume to total volume behind piston at the end of the stroke would be $\frac{4}{10}$; the theoretical efficiency would be therefore—

$$E = 1 - \left(\frac{4}{10}\right)^{0.4} = 0.31.$$

This figure shows the portion of the heat energy of the gas that might, according to theory, be used as work upon the piston with this particular arrangement of engine. The actual indicated efficiency was 0.177, being only 57 per cent of the theoretical efficiency. The difference is due to several causes of loss, such as the heat rejected to the exhaust, walls, leakage, &c.

Loss of Heat to Cylinder and Piston.—The gas on explosion loses heat to the cylinder walls. This corresponds in a measure to initial condensation in a steam-engine, and is affected in extent by the ratio of surface to volume enclosed $\left(\frac{S}{V}\right)$ as well as by the time of exposure. The portion of the loss due to this cause is smaller in large cylinder engines than in those with small cylinders. The loss of heat to the cylinder walls generally figures in heat balance-sheets at from 40 to 50 per cent of the heat units given to the engine, but it is still a matter occupying the attention of gas engineers whether this can rightly be considered as all due to such a simple cause. The temperature reached by the gas on explosion is usually only about half the temperature that ought theoretically to be obtained. Some of the heat of explosion undoubtedly goes direct to the cylinder walls, some does not appear as temperature due to a supposed increase of specific heat of the mixture at the higher temperatures, some of the heat that has gone to the cylinder wall is carried away by the jacket cooling water, and the rest reappears in the gas at the later portions of the stroke, as is evidenced by the curve of the diagram in large engines rising above the adiabatic curve; this latter can only happen when heat is added to the expanding fluid. Further, some of the heat returning *as if* from the cylinder walls is due to the following cause.

Supply of Heat Never Added Instantaneously.—Complete explosion is said to have taken place when maximum pressure has been reached, but combustion has usually not been completed at this time. The continued combustion adds heat to the expanding gas, and so raises the curve above the adiabatic, provided the loss to water jackets is not more rapid than the gain due to this "after burning". In large engines the ratio of surface to volume is often so small that the loss of heat to the jackets is slower than the gain of heat from the "after burning", and the expansion curve rises above the adiabatic curve. In steam-engines the heat is not added all at the highest temperature, but at a temperature which increases as the water is heated from feed temperature to the temperature corresponding to the admission pressure, and afterwards at the constant temperature of admission. Thus an analogy between the steam-engine and the gas-engine in this respect may be drawn.

Diagram Factor.—Admission and expulsion of the gas to and from the cylinder cannot be accomplished without loss of pressure; thus to an extent the "diagram factor" enters into the problem of gas-engine performance as well as of steam-engine performance. It should be noted, however, that "clearance" in a gas-engine is not on the same footing as clearance in a steam-engine. There is no "cut-off" after explosion in the gas-engine, so that clearance controls the number of expansions in the gas-engine, but does not necessarily do so in the steam-engine.

Gain of Heat by Gas Entering Cylinder.—Gas-engines are generally worked with the cylinder as hot as is consistent with safety, so as to reduce the loss of heat to the cylinder walls to a minimum. It follows from this that the incoming mixture will become heated above the temperature of the air. For Otto-cycle engines this heating of the mixture before compression does not affect the theoretical efficiency, but practically it is a cause of loss, because the hotter the mixture before ignition the greater will be the heat of explosion, and the greater the resulting loss to the cylinder walls. With some engines not following the Otto cycle, heating of the charge reduces even the theoretical efficiency. There is no exact equivalent to this cause of loss to be found in the steam-engine.

Summing up the losses as detailed, it may be said roughly that of each 100 British thermal units heat energy supplied to the engine, 40 British thermal units are lost due to the passage of heat to the cylinder walls, dissociation of the gas at the higher temperatures, and the various complex phenomena of explosion and combustion. Of the remaining 60 British thermal units in the case of the Otto-cycle engine mentioned on p. 167, 31 per cent is theoretically obtainable as work on the piston, *i.e.* 18.6 British thermal units should figure as work, and the remaining 41.4 British thermal units would be lost by radiation and carried away by the exhaust gases.

A good steam-engine indicating 22.26 I.H.P. gave a consumption of 12.74 lbs. per I.H.P. hour, which was equivalent to 237 British thermal units per minute. Since 42.4 British thermal units per minute are the equivalent of 1 I.H.P., this engine gave an absolute efficiency of 17.8 per cent. The theoretical heat supply for a perfect engine working under the same conditions would be 150 British thermal units per minute, *i.e.* 63 per cent of the theoretically possible was obtained. Putting the cases of the 20 I.H.P. gas-engine and the 22 I.H.P. steam-engine in parallel columns, an idea can be formed of the relative efficiencies of the two engines.

Gas-Engine.		Steam-Engine.
17.7	Work on piston	17.8
40.0	Lost to jackets, dissociation, &c.	—
42.3	Lost by radiation, &c., and carried to exhaust	82.2
100.0		100.0
17.7	Absolute efficiency	17.8
31.0	Efficiency of perfect engine	28.27
57.1	Relative efficiency	63.0

To carry the comparison down to the heat energy of the coal used, the efficiencies of gas generators and steam-boilers must be compared. Taking the Mond producer, which is able to work with bituminous coal and slack, Mr. Rollason, in a paper read before the Glasgow and West of Scotland Technical College Scientific Society, gives the efficiency of gas generation as 84 to 86 per cent. The late Mr. Bryan Donkin gave it as an average of 70 per cent. For steam-boilers the efficiency ranges from 50 per cent in bad boilers to 80 per cent in good boilers. Taking, therefore, an average figure of 78 per cent for a Mond producer under normal working conditions, and 65 for a steam-boiler, the figures have to be modified as below:—

Gas-Engine.		Steam-Engine.
13.8	Work on piston	11.57
31.2	Lost to jackets, &c.	—
33.0	Exhaust loss, &c.	53.43
22.0	Producer or boiler loss	35.00
100.0	British thermal units from coal	100.00

It is true that larger gas-engines, of from 400 to 600 I.H.P., have given better efficiencies than the above, and figures of 20 to 24 per cent efficiency have been recorded, but it must be borne in mind that large steam-engines also give better results than the above, though probably 20 per cent is about the limit to be expected.

CHAPTER XV

DESCRIPTIONS OF OTTO-CYCLE ENGINE AND DOUBLE-ACTING KÖRTING ENGINE

Westinghouse Three-Crank Otto-Cycle Engine.—Fig. 1353 shows a section through one cylinder of the engine, all three being exactly alike. The cylinders are mounted on an enclosed crank-case, which forms the base or frame of the engine, and serves as a receptacle for the oil which lubricates the cranks and upper journals of the connecting-rod. The engine is single-acting. A simple trunk piston, carrying a hardened steel wrist-pin, which is linked to the crank by the connecting-rod, comprises the mechanism for transforming the intermittent pressures on the top of the piston into rotary motion at the shaft. The valves and igniters perform their functions once for every two revolutions of the crank, and are operated by cams mounted on shafts running only half as fast as the main shaft. Referring to fig. 1353, A is the shaft which carries the exhaust-valve cams, and is driven from the main shaft by gearing. Each exhaust cam works against a roller carried on the free end of the guide-lever G. The exhaust-valve E has a long stem projecting downward and resting on a hardened steel plate on the upper side of the guide-lever G. The spring surrounding the stem serves to hold the exhaust-valve to its seat and the stem in contact with the guide-lever. From the exhaust cam-shaft A a horizontal shaft with bevel-gears leads to the opposite side of the engine, engaging with a vertical shaft which in turn drives the upper cam-shaft B. Incidentally, the vertical shaft carries the governor. The upper cam-shaft carries two cams for each cylinder. One engages against a roller on the end of the horizontal lever C. As the throw side of this cam comes uppermost, the opposite end of the lever C depresses the stem of the inlet-valve J, opening the latter for the admission of the mixture of gas and air. A spring on

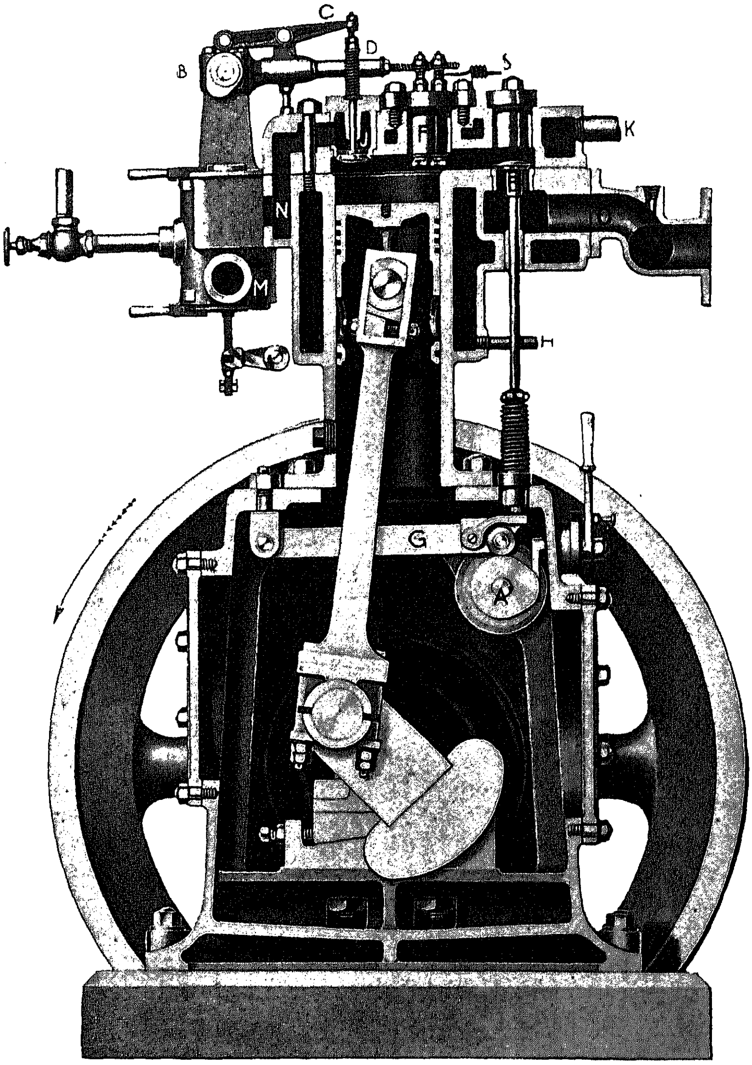

Fig. 1353.—Westinghouse Gas-Engine

the stem of the inlet-valve furnishes a means for closing it and keeping the cam and roller always in contact with each other. Immediately adjacent to the inlet-valve cam is the igniter cam, which at the proper instant operates a horizontal plunger working through the guide D to break the electric current through the wire S at the terminals of the igniter F. Cooling water enters at H and is discharged at K. The gas and air enter the mixing valve chamber M by separate inlets, and the mixture is conducted through a distributing chamber to the port N leading to the cylinder head in which the inlet-valve is placed. The engine is started by means of compressed air. The air supply is furnished by a small compressor, and is stored in iron tanks under a pressure of 250 lbs. to the square inch. On the end of the upper cam-shaft on the engine is a screw with a milled head, which, on being given a few turns to the left, renders the admission-valve of one cylinder inoperative for the time being. On the back of the crank-case is a short lever, which, on being moved to the right as far as it will go, throws out the regular exhaust cam and throws in a supplementary cam, which keeps the exhaust-valve open during *every* up-stroke of the piston, instead of *every other* stroke as when the engine is in regular operation. A pipe leads from the air-storage tank to the starting-valve, which is attached to the end of the crank-case, and thence through a check-valve into the cylinder in which the valve functions have been altered. This valve is actuated by a cam on the end of the lower cam-shaft (fig. 1354), where it projects through the end of the crank-case, and is so timed as to open each time the piston starts on its downward stroke. It will be readily seen that one cylinder is thus converted into a compressed-air motor without disturbing the rest of the engine. The engine being set with the corresponding crank a little past its upper centre, the air and gas supply properly adjusted, and the stop-valve in the air-pipe opened, it starts off and continues to run on air pressure until explosion takes place in the other cylinder or cylinders. The air supply is then shut off, the inlet and exhaust valves put back in their normal working condition, and the engine is in full operation. The starting-valve

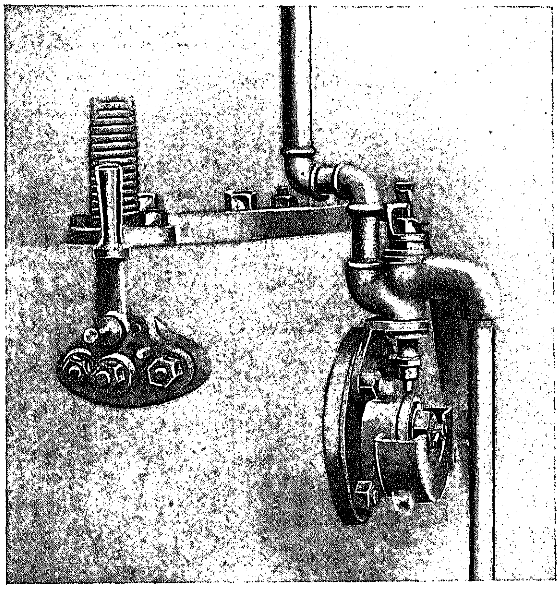

Fig. 1354.—Westinghouse Starting-Valve

may be disengaged from its cam when not in use by pulling up on the stem and giving it a quarter turn in either direction.

The action in the cylinders when operating by explosion is as follows:—On the first down-stroke air and gas flow into the cylinder through the inlet-valve. The next up-stroke compresses the charge. The second down-stroke, which is the power-stroke, expands the burning mixture which has been fired just at the top of the stroke. The second up-stroke expels the products from the cylinder. This is the usual Otto cycle. There being three cranks, each with a cylinder operating it, there are three impulses on the generator every two revolutions.

Mather & Platt "Körting" Gas-Engine.—The Körting engine is of the double-acting two-cycle type. This engine is therefore built on exactly the same lines as an ordinary double-acting steam-engine. Fig. 1355 gives a general view of a 500-B.H.P. engine. Each side of the piston receives an impulse every alternate stroke, *i.e.* there are two impulses per revolution. At the front end the piston-rod is carried through a stuffing-box fitted with metallic packing, and is fitted to an external cross-head in the usual way. The outside cross-head, while permitting of free lubrication and cool running of the pin and guides, also does away with wear and tear of cylinders and pistons of the trunk type. The piston is long, affording a good wearing surface, and the front gland together with the main guide ensure perfectly true and smooth running inside the cylinder. Alongside the power cylinder are placed two charging pumps, one for air and one for gas, which pumps play an important part in the cycle of the engine, as will be understood by referring to figs. 1356, 1357, 1358. When the piston P is on the forward dead-centre, as shown, it is evident that the exhaust-ports S round the centre of the working cylinder are uncovered, their opening having commenced when the face of the piston was in the position represented by "*d*" on the indicator diagram. At that moment

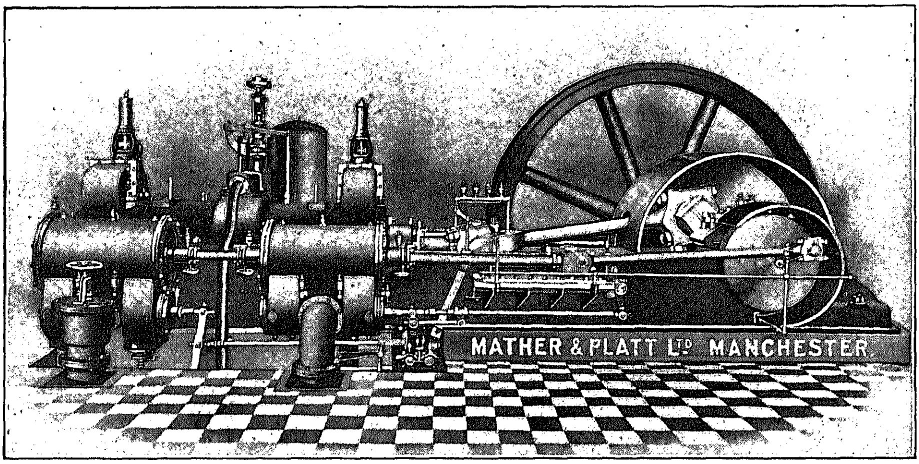
MATHER & PLATT L^TD MANCHESTER.

Fig. 1355.—Körting Gas-Engine

the cylinder pressure, still apparent from the expansion curve, falls to atmosphere; at the same time the admission-valve E is opened by its cam, and a scavenging charge of pure air, at a pressure of 5 or 6 lbs. above atmosphere, is sent into the cylinder by the air-pump, effectually clearing out the products of combustion. The gas-pump, whose discharge has been delayed, now sends a supply of gas into the cylinder, which discharge, mingling with that of the air-pump, forms the combustible mixture necessary for the power-stroke. The exhaust-ports S are then closed by the return stroke of the power piston P, the mixture is compressed in the ordinary way, ignited electrically at two distinct points to ensure complete firing, and the impulse is transmitted to the piston. The same cycle is repeated at each end of the piston alternately, the air-pump thoroughly sweeping all burnt products through the exhaust-ports S, and thus entirely preventing the ignition of the incoming charge by contact with the hot gases remaining in the cylinders.

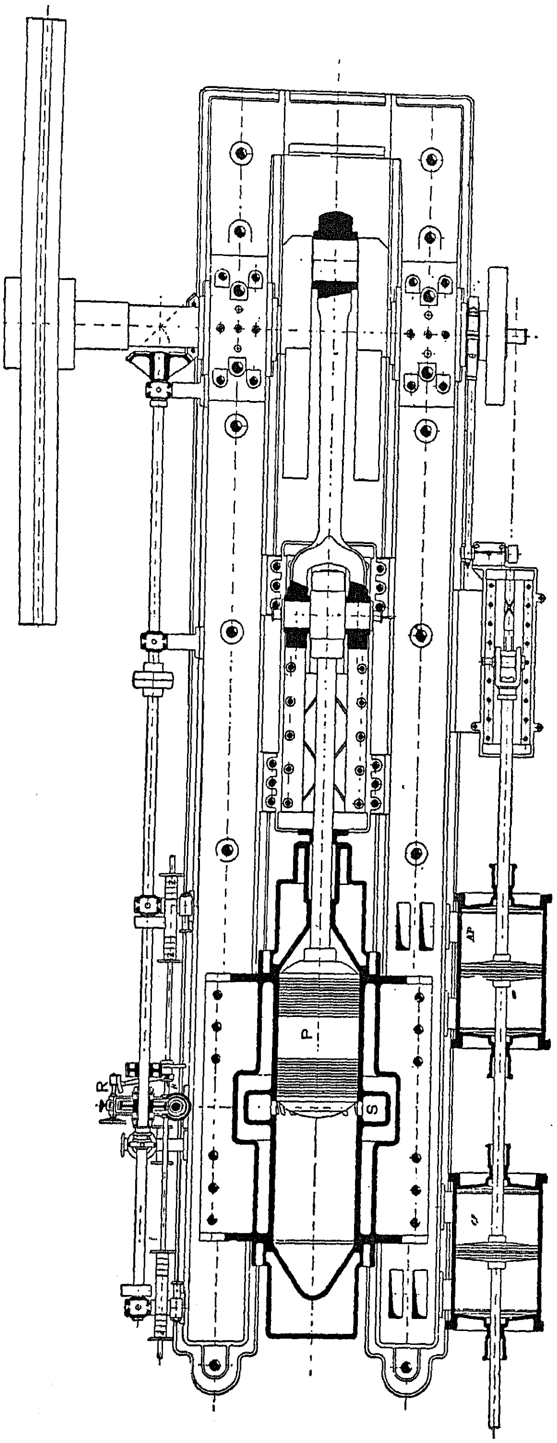

Fig. 1356.—Körting Gas-Engine

Regulation is effected by altering the quantity of the mixture admitted to the power cylinder R. During periods of light load the valve admitting gas to the cylinder is kept closed for a longer period than during heavy load, and *vice versa,* so that the charge is proportioned to the work done though the quality of the mixture at the points of ignition is maintained practically constant, thus ensuring full ignition. The mixture is fired, as already stated, at two independent points at each end of the cylinder, and an arrangement is provided whereby the period of ignition can be made earlier or later by hand while the engine is running. Weak gases, such as blast-furnace gas, require to be ignited earlier than "producer" gas, and the latter, in like manner, earlier than "town" gas.

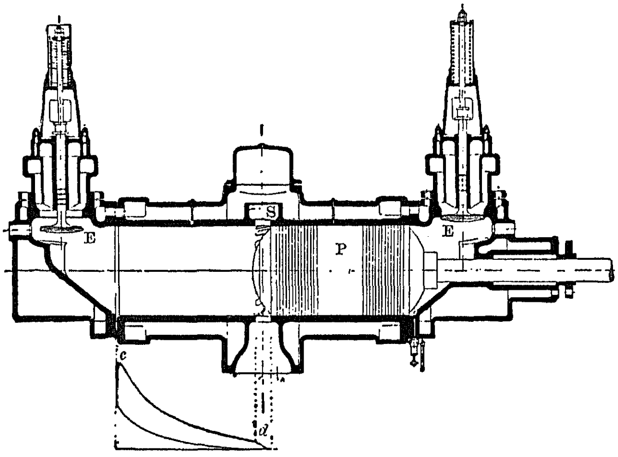

Fig. 1357

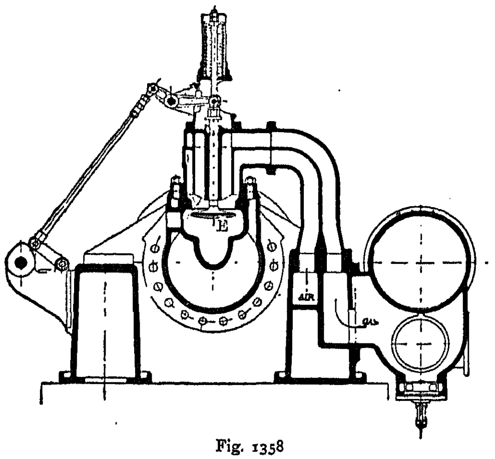

Fig. 1358

The cooling of the engine is very perfectly carried out. In the first place, the power cylinder is completely water-jacketed, including the cylinder heads, metallic-packing stuffing-box, and exhaust-port belt. By means of rocking pipes, held in water-tight glands, circulating water is carried to one side of the main engine cross-head, from whence it travels through the hollow piston-rod into the hollow piston, and back through a pipe located inside the rod to the other side of the cross-head, and then away. Thus the interior of the working cylinder is kept at a reasonable temperature, and, by means of thermometers placed in suitable positions, it is quite easy to determine the relative temperatures of the cylinder and piston, the latter being always kept a few degrees cooler than the former.

CHAPTER XVI

COST OF GENERATION OF ELECTRICITY BY MEANS OF GAS-DRIVEN PLANT

The following figures of costs of generating electricity by means of gas-driven dynamos will give an idea of the present position of gas-engines in this respect.

A report by Messrs. W. P. Thurmann & Co. on the cost of generation at the Heywood Industrial Co-operative Society, Ltd., where only a small plant is installed and "town" gas used, shows the following figures:—

Cost of Gas per Unit of Electricity Generated.	Oil, Waste, &c.	Wages.	Interest and Depreciation.	Total Cost per Unit. Pence.
d. 0.92	*d.* 0.10	*d.* 0.77	*d.* 1.06	2.85

Output, 17,862 units of electricity.

As regards the cost when the gas is made in a "producer", Mr. W. Langdon has kindly supplied the following figures from the electric-lighting plants under his control on the Midland Railway. As the company is its own consumer, the costs, especially as regards maintenance, are not on all-fours with those given by electricity-supply companies using steam-engines, because the expense of keeping and trimming lamps and fittings, usually borne by the consumer, appears in the costs of running the electric-light plant for the railway-stations. In order, therefore, to give a fair idea of the position of the gas-engine with producer, the costs of the most economical steam-driven electric-light plant on the Midland Railway are also given. The figures are the averages for the years 1900 and 1901 in each case.

LEICESTER STATION. GAS-DRIVEN PLANT, DOWSON GAS. AVERAGE OUTPUT, 180,895 UNITS, OF WHICH 7.7 PER CENT WAS GENERATED BY PLANT USING "TOWN" GAS

Cost of Coal and Water.	Maintenance.		Repairs.		Total Cost Per Unit. Pence.
	Labour.	Material.	Labour.	Material.	
d. 0.503	*d.* 1.209	*d.* 0.469	*d.* 0.049	*d.* 0.015	2.245

As the cost of making the Dowson gas (*including labour*) was 0.581*d.* per unit, and the price paid for the town gas was 1.615*d.* per unit of electricity generated by it, an idea may be formed of what the total cost per unit would have been had all the gas been producer gas.

The most economical plant using steam-engines and boilers was—

HIGHGATE ROAD. WILLANS ENGINES. LANCASHIRE BOILERS. AVERAGE OUTPUT, 953,000

Cost of Coal.	Maintenance.		Repairs.		Total Cost Per Unit. Pence.
	Labour.	Material.	Labour.	Material.	
d. 0.773	*d.* 0.657	*d.* 0.522	*d.* 0.229	*d.* 0.119	2.30

As in the chapter on "Reciprocating Steam-Engines" these costs include repairs and attention to dynamos, switches, and cables, as well as engines and boilers or producers.

The above seems to show that the gas-driven plant, when the gas used is "producer" gas, has a slight advantage over steam-driven plant for continuous-current work.

3. Condensing and Auxiliary Plant

CHAPTER I

CONDENSERS

General.—The object of condensing the exhaust steam from an engine in a closed vessel is to remove the pressure of the atmosphere from the piston face, thereby adding to the mean effective pressure in the cylinder to the extent of 12 to 14 lbs. under favourable conditions. Assuming that in any non-condensing engine the mean effective pressure is 36 lbs. per square inch, and that the back pressure or exhaust pressure be reduced 12 lbs. by condensation, then the new mean pressure will be 48 lbs. The amount of work done by the engine for the same weight of steam will thus be increased by one-third. In modern factory engines with a steady load the mean pressure, referred to the low-pressure cylinder, should be 40 to 45 lbs. Of this about 14 lbs. is due to the condenser, which therefore effects an economy in the ratio of $\frac{40}{40-14}$ to $\frac{45}{45-14}$, or from 35 to 31 per cent. In electric traction work, where load factors are from 0.2 to 0.8 according to the number of the cars run from the station, the mean effective pressure will often be very much below the economical load of the engine. In such cases the proportion due to the vacuum is considerable, and economies upwards of 40 per cent may be estimated as between a condensing and a non-condensing engine. The atmospheric pressure at sea-level is 14.72 lbs. per square inch, which is equal to a head of 34 feet of water, or to a column of mercury 30 inches or 760 millimetres high. If, in fig. 1359, C is the cylinder of a steam-engine, and B is the condenser into which flows a jet of cold water from a reservoir R, this cold water will condense the steam as it enters the condenser when the exhaust-valve E is opened.

From the condenser a pipe is carried down to a distance H, which need not exceed 34 feet. The end of the pipe is turned up and immersed in the discharge-tank T. This arrangement will form an excellent condenser, but there are two objections to it. First, the water supply and the position of the condenser, 34 feet above a convenient discharge point, will rarely be found. Secondly, the action will soon cease, because the condenser will become full of air or other gases. These gases reach the condenser in various ways. The feed-water pumped into the boiler carries gas in solution, and this comes out of solution by heat and passes forward with the steam. Similarly, the injection water carries in its share of dissolved air, which comes out of solution under the joint influence of warmth and reduced pressure. Finally, there is the considerable volume of air which leaks through bad exhaust-joints, porous castings, and badly packed glands. To deal with these gases an air-pump is required of such capacity as will enable it to remove the gases sufficiently to maintain their pressure at a point as low as the pressure of water-vapour at the temperature of the condenser. The air-pump is usefully employed also to remove the water from the condenser at the same time. We are thus able to dispense with the arrangement of fig. 1359, and apply condensers under any circumstances. There are few non-condensing engines in which the exhaust pressure is not above the atmospheric line. The economy due to condensing is thus measured not by the amount of "vacuum" below the atmosphere, but by the total reduction of back pressure below the exhaust line of the engine when run non-condensing, *i.e.* by the amount *p* in fig. 1360.

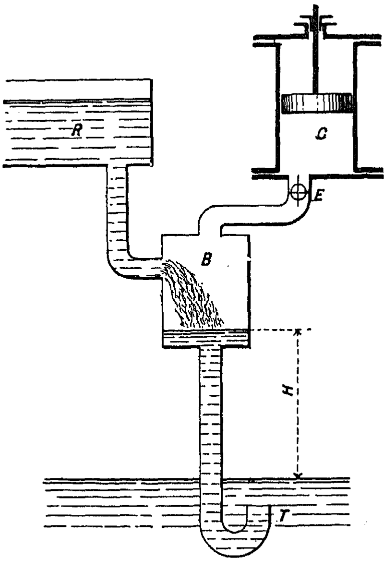

Fig. 1359

A table of the properties of saturated steam shows that a pressure of 1 lb. absolute corresponds with a temperature of nearly 102° F., or a vacuum of 13.72 lbs., or 27.96 inches of mercury. As in practice it is rarely found that a vacuum better than 12.72 lbs. can be secured, we need hardly provide for a condenser temperature much below 126°, which corresponds with an absolute pressure of 2 lbs. or a vacuum of 12.72 lbs. There are two good reasons for not making the condenser too cold. (*a*) More water is required, and this absorbs power in getting rid of it. (*b*) If the feed is drawn from the condenser discharge the heat is valuable both as heat to save fuel, and because it is not proper to feed an economizer with water much if at all below 100° F. Probably a condenser temperature of 100° to 110° will give satisfactory results all round. Another objection to excess of water run through a jet condenser is the further volume of air it brings with it.

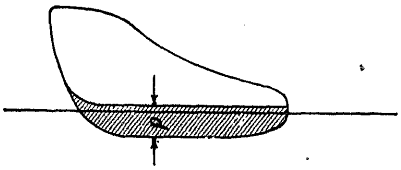

Fig. 1360

It is a simple matter to calculate the amount of heat to be dealt with by a condenser. One pound of exhaust steam measured from the ordinary condenser mean temperature of about 100° to 120° contains 1050 heat units or less. A convenient round number is 1100 units. If the steam is cooled to 100° the injection water will be raised to the same temperature. If this water is added at 50°, each pound will absorb approximately 50 heat units. Then $1100 \div 50 = 22$. In brief, the injection water would require to be provided to the amount of twenty-two times the weight of steam to be condensed in this case.

Calling this ratio R, we can always calculate it by the formula,

$$R = \frac{W}{w} = \frac{1100 - T}{T - t},$$

where W = weight of injection water,
w = „ steam to be condensed,
T = temperature of condenser,
t = „ injection,
1100 = heat units in each pound of steam above condenser temperature.

Then also $T = \frac{1100 + Rt}{1 + R}$, which shows that this expression follows the condenser temperature. Within reason the mean effective cylinder pressure increases as T diminishes, and good practice finds a balance at 100° to 110°, and 25 lbs. to 27 lbs. of water per pound of steam to be condensed.

Condenser Volume.—The size of a condenser depends primarily upon the water and air put into it, and the capacity and speed of the air-pump. The volume of air to be dealt with has a threefold origin. There is first that which comes in with the injection water, and usually about one-twentieth the volume of the water. About five times the quantity is considered to get in at the cylinder glands, or in all, three-tenths the water volume. The actual volume occupied by the air at the condenser pressure will be from ten to fourteen times this according to the quality of the vacuum, so that the total air volume to be pumped will be about 3.6 times the injection volume, and the mixed air and water will be 4.6, or, say, five times the injection, as the minimum capacity of condenser measured from the volume of injection water introduced between consecutive strokes of the air-pump. Thus it appears that fast-running pumps, or multiple-barrel pumps, which amount to the equivalent of fast-running, require less condenser volume than slow-running pumps. In practice these facts are rarely considered, it being usual to give a condenser volume of at least a quarter of the last cylinder, and even as much as half that volume.

Ejector Condenser and Air-Pump.—A form of condenser without moving parts is that known as the ejector condenser. It depends for its action upon the velocity with which steam will flow into a space at less pressure.

Steam obeys the laws of liquid flow within certain limits. The velocity of outflow of a liquid is given by the formula $V = \sqrt{2gh}$, where V is the velocity in feet per second, h is the head in feet, and g is the acceleration due to gravity, or 32.2 in foot-pound units. Approximately $V = 8\sqrt{h}$.

For steam, the height h depends upon the density. At 3 lbs. absolute pressure this is about $\frac{1}{7380}$ of water; so that if the net difference of pressure is 2 lbs. in the ejector, or equal to, say, 4.6 feet head of water, the equivalent steam column is 33718 feet = 4.6 × 7330, and by the formula $V = 1472$. A simple formula for velocity of steam is $60.2\sqrt{T}$, where T is the absolute temperature of the steam. This is a general formula. Steam of any pressure flowing through an opening into any other pressure less than three-fifths the initial pressure is said to have a velocity of 888 feet per second, but probably this is too low a figure. In any case the velocity of flow is very great, and when such a flow of steam is mixed with a jet of water, the combined momentum will be the mean of the two streams. If the weight of steam be one-twenty-fifth that of the water, the combined jet will have a velocity one-twenty-sixth that of the steam, or, in the present instance, not less than 34 feet per second. If the higher velocity for steam be correct, the combined velocity will be about 60 feet per second, a velocity that corresponds with a head of about 60 feet of water, or a pressure of 25 lbs., so that there would be a margin of 50 per cent for efficiency to give a vacuum of 12½ lbs. The ejector condenser is, in fact, only an application of the same principles as apply to the injector, and over 12 lbs. of vacuum may be secured by their means. Where the water will flow to the condenser, the cost of working an air-pump is saved, because the ejector works by the heat energy of the exhaust steam.

Generally no air-pump acts so effectively as the vertical single-acting type, which enables the whole of the valves to be kept sealed by water, and thereby assists to prevent reflux of air; but the ejector condenser gives very good results, especially when its supply of water approaches with considerable velocity, as from a head of 15 feet, or from a centrifugal pump.

With jet condensers, placed not more than 20 feet above the source of water, the injection will be raised to the condenser by the action of the vacuum. Otherwise the water must be lifted by a pump. In surface condensers the water must be supplied either by gravity or by a pump, and it is very convenient, especially in electrical works, to employ pumps of the centrifugal type, which are so easy to drive direct by an electric motor.

The capacity of the circulating-pump should be such that it will give the necessary amount of condensing water required, or about twenty-five to thirty times the feed-water in case of jet condensers, up to as much as seventy times for surface condensers, the efficiency of which depends upon the rate of passage of heat through the walls of the thin tubes of brass of which condensers are usually made. Experimental data as to the efficiency of condenser tubes are not plentiful, but practice has fixed an ordinary ratio of 10 lbs. of steam condensed per hour per square foot of tube surface. It is claimed for the Row indented tube that the efficiency is double that of plain tubes, owing to the turbulent flow of water that these tubes cause.

The Jet Condenser. — The jet condenser is a very simple structure. Fig. 1361 shows a common form of condenser. Steam enters at S. Water enters at W, being forced in by a pump or by the external atmospheric pressure. The base of the condenser communicates with the air-pump. It was once the custom to admit water through a perforated rose, but this was discarded as not being necessary, and a simple inlet only was provided, though it is probable that the eddies in the jet caused by the passage of the water through the inlet valve will so break up the stream as to approximate to the conditions given by a perforated pipe. The effect of the jet appears to be sufficient instantly to condense the entering steam. These condensers are made with a volume of one-fourth to one-half that of the low-pressure cylinder. In old-fashioned engines, which obtained a vacuum of over 14 lbs., it was often found necessary to provide an air inlet-pipe to the condenser to ease the shock when the air-pump delivered its charge. Some air was necessary to do this, and in well-kept engines enough air was not always provided by leakage, and had to be intentionally given.

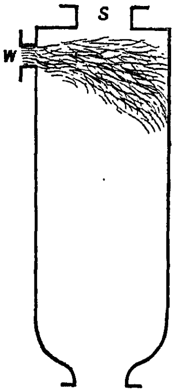

Fig. 1361

Some engineers consider that the surface condenser is only suitable where the water supply is dirty, and thereby the condensed steam can be saved pure and clean if satisfactorily purged of oil. But where clean water can be had, the jet condenser is still much used. In the counter-current condenser of Balcke, for particulars of which we are indebted

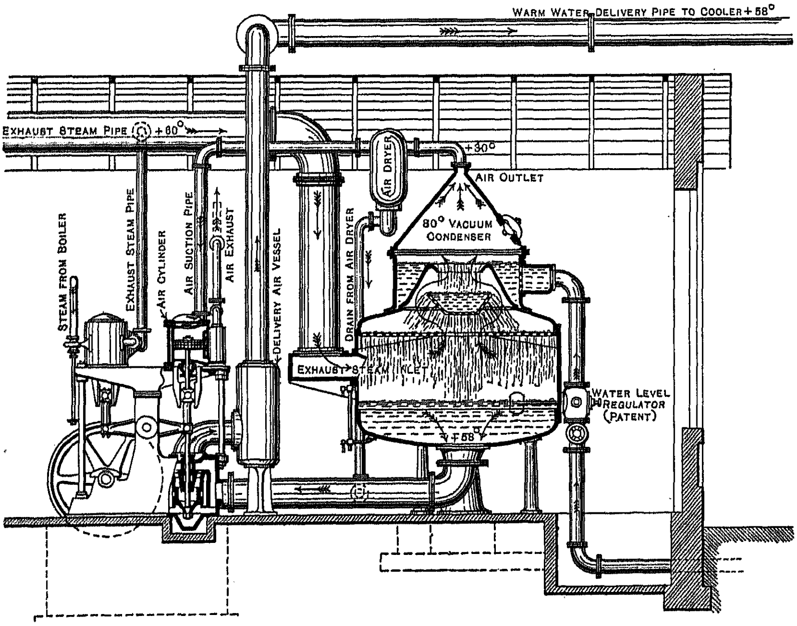

Fig. 1362.—Jet Condenser

to Mr. Koppel, of 27 Clements Lane, London, the exhaust steam enters the condenser at about middle height. The circulating water enters above, and is caused to fall in a shower, as shown in fig. 1362. The air-pump is double, and consists of a lower water-pump, which abstracts a fixed volume of water from the base of the condenser, and of an upper dry air-pump, which abstracts air from the upper space. A float-valve regulates the injection exactly to the amount of the water abstracted by the water-pump. Thus the hottest steam comes in contact with the hottest water, and the air at the colder upper end is reduced to its minimum volume. The boiler feed is drawn from the lower and hotter end of the condenser and passed forward to the feed-heater. The lower pump delivers the water to the top of the cooling-tower or cooling-pond, whence it is returned cooled to the suction-well of the condenser. By the arrangements here shown the injection water is supplied in known or desired quantity, which is difficult to arrange with plain jet condensers by means of the ordinary injection-valve, for this admits less water when it should admit more, owing to the reduced vacuum drawing the water less powerfully. The float-regulated valve, being large enough for all purposes, will admit all the water necessary even at a poor vacuum, for as soon as the supply falls a little short, the water-level in the condenser base is lowered, and the float falls and opens the injection-valve wider.

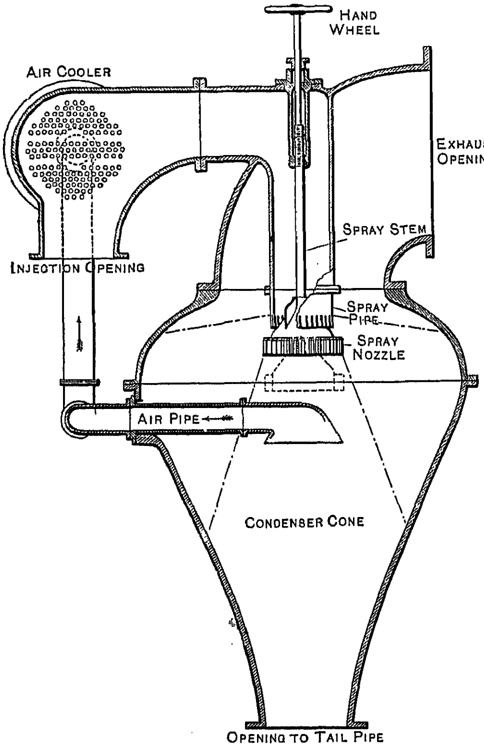

Fig. 1363.—Section of Worthington Central Condenser

Barometric Condenser.—In the barometric condenser of the Worthington Company the condenser (fig. 1363) is placed at a height of not less than 30 feet, and the jet of water for condensing is forced up by a pump, the duty of which is merely that due to the difference between 30 feet and the head equivalent to the vacuum. A separate pump is provided to draw off the air, which passes through a tubular air-cooler, by which the injection water also passes on its way to the jet, thus reducing the air in bulk and rendering the air-pump more efficient. This barometric condenser is the practical application of the principles of fig. 1359, already dealt with. This condenser is particularly suitable where an ample stream of water from a higher point is available. It will maintain a fair vacuum without an air-pump, the air being entangled with the water column and drawn away. The gravity pipe should be as small as consistent with dealing with all the injection water, for the velocity of flow assists in the withdrawal of the air. The ejector condenser may be placed to act with a barometric discharge. The barometric condenser may also be combined with a surface condenser, as in fig. 1364, the air-pump being arranged to draw off air both from the main down-flow and from the surface condenser. The water discharged from the surface condenser may have its own separate down-pipe or may go through the air-pump. With separate down-pipe it may pass through an oil-filter and forward to the feed-pump in the same closed circuit.

Surface Condensers.—These act on the principle of providing water-cooled pipes, between which and by contact the steam is condensed. Fifty to seventy times the weight of steam condensed is made to flow through the tubes of the condenser.

Stanton[1] has shown that the best results are obtained with small tubes of great length placed vertically and with down-flowing water. Professor Reynolds showed that water has a turbulent flow when its velocity exceeds the critical velocity $V = \frac{P}{84.7\,D}$ feet per second, where D is the diameter of a tube in feet, and P is the value of an expression based on the temperature Centigrade T, and $P = (1 + 0.0336\,T + 0.000221\,T^2)^{-1}$. Actually, where T is changing all the time, turbulent flow will occur at lower velocities. Turbulent flow adds greatly to the efficiency of heat absorption.

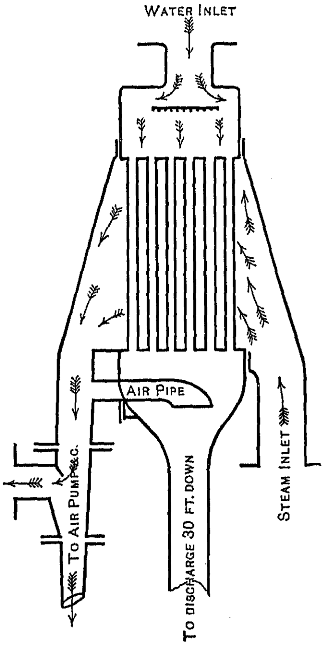

Fig. 1364

The amount of surface necessary in a condenser is usually from 1 to 2 square feet per I.H.P. A better rule is to allow, say, 10 lbs. of steam per hour per square foot of tube surface. Tubes are usually $\frac{5}{8}$ to $\frac{3}{4}$ inch diameter and $\frac{1}{20}$ inch thick, but it would appear that less diameters would be better, and even $\frac{3}{8}$ and $\frac{1}{4}$ inch have been suggested.

In practice, the tubes are usually horizontal for convenience of access, and the water often flows in one direction through half the tubes and in the reverse direction through the other half. The steam should first meet the hotter tubes. One of the difficulties to contend with and to guard against is the flow of the exhaust steam by too short a route through the condenser. Unless provided with suitable baffle-plates to distribute the steam, this is apt to take a short circuit to the outlet without passing over sufficient tube surface. In fig. 1365 is shown a surface condenser of the

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[1] *Minutes of Proc. I. C. E.*, vol. cxxxvi, part ii.

Wheeler Condenser Co. of London, together with air and circulating pumps complete.

The perforated baffle-plate is seen immediately below the exhaust inlet. The water-cooled tubes are arranged on the Field system, water entering from the outer water chamber and passing through the inner tubes, and returning by the space between the inner and outer tubes. These complete plants are driven, as in fig. 1365, by a central steam cylinder. Faulty action is often due to the cockling of the india-rubber valves of the air and circulating pumps from the heat or the action of the mineral cylinder oil. If dermatine valves are employed this trouble will cease.

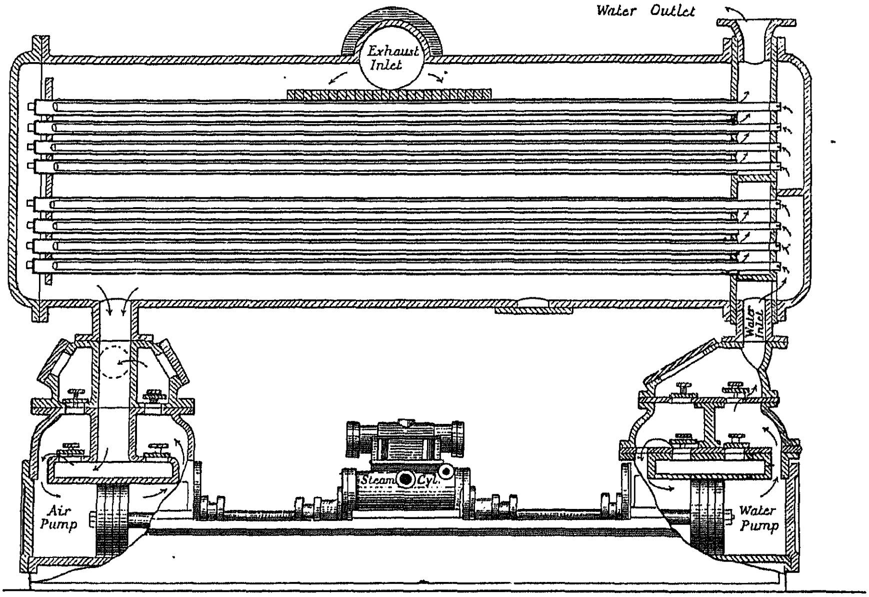


Fig. 1365.—Wheeler Condenser

Condensers of this class are also worked by means of vertical air-pumps and centrifugal circulating-pumps, and the surface condenser may be combined with a feed-heater, the heater tubes coming next the entering steam. In figs. 1377 and 1379 the condenser tubes are seen to be plain and in two sets, the circulating water passing first through the lower set and next through the upper set, as shown by the arrows, the feed, of course, being quite separate.

Owing to the constant movement of condenser tubes under the varying temperatures to which they are exposed they cannot be rigidly held in their tube plates. In the type of fig. 1365 the tubes may be rigidly fixed at one end, because the closed ends of the tubes are perfectly free to move.
the type of fig. 1377 one end may be screwed into its plate, but the end must have a packing and gland ferrule. A little soft cotton and linseed-oil or tallow makes a fair packing for condenser tubes.

Many condensers have simply thin brass tubes packed thus at both ends, and sometimes wooden ferrules are used for packing the glands.

Some makers use condenser tubes as large as 1⅝ inch diameter, so that they may make their condensers long. It is better to arrange for a central supporting diaphragm and small tubes. A central support will quadruple the strength of a beam and will multiply the stiffness by ten, and condenser tubes are simply beams.

Almost invariably the surface condenser is made with horizontal tubes, but it would probably be improved if they were set vertically, the steam entering at the lower end, and air being withdrawn at the top end by a pipe leading to the air-pump entry. The circulating water should descend if only one direction of flow is arranged for, otherwise the arrangement should be such that the entering steam meets first the hottest water. It is by no means necessary that the cooling water should pass through the tubes. Surface condensers may be arranged so that the steam passes through the tubes and the cooling water surrounds them. The same precautions must be taken to prevent the water from travelling by too short a circuit to the outlet point. This arrangement is a usual one with water-supply companies, who pass the whole of the water, pumped by an engine, through the surface condenser of that engine. The large supply of water results in a very small addition to the temperature being made.

With water containing scale-forming matter the tubes of a condenser will become encrusted on the water side. They may be cleaned by means of a 20-per-cent solution of the commonest hydrochloric acid, which will dissolve the crusts so that the tubes may be washed out. If the solution can be kept circulating the cleaning will be more even and rapid. When the process is complete, the evolution of carbonic acid gas from the lime carbonate scale will cease. The acid may then be run off, and followed by a circulation of alkaline water to neutralize any remaining acid. Sulphuric acid will also destroy carbonate scales, but will be less efficient where any sulphate scale is present. It is found that where there is scale that can be attacked by acid, the acid attacks the scale before it attacks the metal of the condenser. The process is thus safe to adopt whether scale is inside the tubes or external to them.

The Ejector Condenser.—One form of the ejector condenser, which was invented, we believe, by Alexander Morton of Glasgow, as made by Ledward & Co., is shown in fig. 1366. Its principle has already been described, and more need not be said, except that it is suitable for use where the water supply is large and comes from some elevation. Otherwise a centrifugal pump is used to force water through the instrument. The makers rate these condensers as requiring about twenty-seven times as much water as steam condensed for the small size of 1½ inch, having a capacity of 200 lbs. of steam per hour condensed up to the large 18-inch size capable of condensing 36,000 lbs. of steam per hour. Water may be pumped to the condenser at a cost of 3 per cent of the economy gained by condensing, or about 1 per cent of the main engine power.

Surface Condensation.—In the case of surface condensation the injection water and its contained air do not affect the condenser volume, nor are they dealt with by the pump. The gland leakage remains, and the air volume is about five-sixths of that in the injection condenser, or three times the volume of water required for a jet condenser. In practice the condenser volume will be about three-fifths to four-fifths the volume of a jet condenser, in addition to the tube volume. The volume of circulating water to be employed in a surface condenser is found to be from two-and-a-half to three times that required in jet condensing, owing to the resistance of the tubes to the transfer of heat, and to the difficulty of securing that the steam shall travel over all the tube surface.

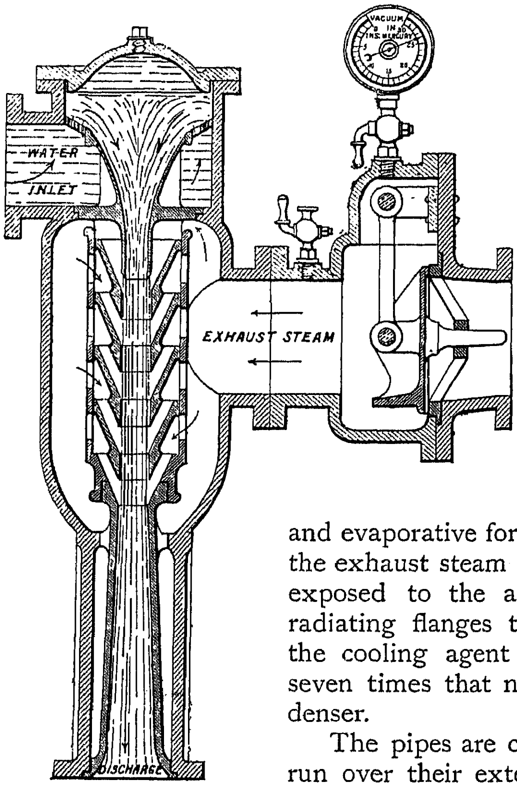


Fig. 1366.—Ejector Condenser

As air is the great cause of poor vacuum it is important that the piston-rod glands and other glands exposed to vacuum should be in good order and well packed, and that all pipe joints should be tight, and all surfaces painted to close up pores. Chaplet holes are a frequent cause of air leakage, especially in long injection pipes.

This point is of special importance in the atmospheric and evaporative forms of condenser. In the former the exhaust steam is passed through a set of pipes exposed to the atmosphere and provided with radiating flanges to facilitate cooling. As air is the cooling agent the surface required is about seven times that necessary in a water-cooled condenser.

The pipes are cooled by thin streams of water run over their exterior surface. The evaporation of this abstracts heat from the steam inside. It is found that for each pound of steam condensed, about two-thirds or three-fourths of a pound is evaporated. These condensers are practically useless in a fog when the air is already laden with moisture. They act best in dry, breezy weather.

A coil of gilled pipes, containing twenty pipes placed ten high and two in length, will condense 700 to 1000 lbs. of steam per hour. Such condensers, when tight and working well, will give a vacuum of 26 inches of mercury.

Condenser Calculations.—In calculating condenser plant, especially for electric traction work, it is better to take no heed of any statement of horse-power, but to ascertain the mean load on the station in kilowatts, and, from known steam consumptions, find the actual weight of steam to be condensed, remembering that the coal consumption per kilowatt of output may range from 3 lbs. per kilowatt-hour in a large condensing station with a good load factor, up to 12 or 14 lbs. for small stations with poor load factors.

General Arrangement.—Condensing plant may be arranged either in separate units for each engine, or as an independent central establishment to which each engine delivers its exhaust; or a combined system may be used where each engine drives its own air-pump and exhausts air from a central condenser. The first system is claimed to be advantageous in that the air-pump may be set to work before starting the main engine, thus providing a ready-made vacuum to start with. Also, the central air-pump may vary its speed to suit the number of main engines at work, and it need hardly be mentioned that the speed of the central air-pump should always be reduced until the vacuum-gauge shows slight backward movement, for it is a waste of power to keep the air-pump running fast simply to pump vapour without increasing the vacuum. Where a condenser is very hot no increase of air-pump velocity within reason will give a good vacuum. Where a condenser is cold the cause of bad vacuum must be sought in leakage of air.

The direct connected air-pump has the advantage that the main engines will work the pump more economically than the smaller and less economical engines of the separate plant, but there may be a sufficient use for the steam from the separate plants fully to warrant their use, as in feed-heating, or they may exhaust to the casing of the low-pressure cylinder of the main engine.

The combined system with separate air-pumps and central condenser has the advantage that the full condenser effect is available for a portion only of the main engines when all are not at work, while the air-pump capacity is automatically fixed by the number of engines at work. It also provides a ready-made vacuum for starting all engines except the first. The second system also can be arranged by means of small equalizing pipes to get up a vacuum in each newly-started engine except the first. In what follows a few types have been described and illustrated to show the various forms and combinations in which condensing plant may be arranged. Further and detailed particulars would far exceed the writer's allotted space, but can be obtained from the makers of condensers, air-, circulating-, and feed-pumps, cooling-towers, feed-heaters; only a few of whose names will be found here, but who give attention to some or all of the various appliances. It is not to be assumed that particular apparatus illustrated is necessarily indicated as better than others, the writer's object having been simply to pick his illustrations over as wide a field as possible.

Data necessary for Estimate of Cost.—The points and data for designing condensing plant may be briefly summarized as follows:—

1st. The weight W of steam to be dealt with per hour, whence is found the total heat to be dispersed or absorbed = W × 1100 thermal units.

2nd. The temperature T of water for condensing, the net quantity of which usually will be $\frac{1100\,W}{110^\circ - T}$; 110° being an assumed condenser temperature. For surface condensing double to treble this circulating water is required.

3rd. Area of tube surface in surface condensation $= A = \frac{W}{10}$.

4th. Weight of air to be passed through cooling-towers $= W \times 25$.

Estimates can be obtained for plant and should include a statement of the essential points above named. With the estimates before him, and the conditions of his site, the engineer will then be able to select the system that seems best adapted to his station. The absence or presence of one factor may vary the whole plant. Thus, with space for chemically softening water the feed-heating system may be varied. Space for cooling-ponds will obviate the necessity of cooling-towers. A constant ample stream of water, especially if it falls from above, may warrant the ejector condenser, or at least avoid the expense of circulating-pumps, or enable the barometric condenser to be used either in its jet form or as a surface condenser. It should also be noted that the ordinary surface condenser may be worked on the jet-condenser principle of the circulating water being dealt with by the air-pump, the condensed steam being drawn off by a separate pipe to the hot-water pump, which may be the feed-pump, the air in the surface condenser having an outlet to the air-pump, as shown diagrammatically in fig. 1367. In this way the pure condensed steam may be saved for feed-water, and one pump made to do duty as air and circulating pump. Some of the steam would be lost, being drawn off as vapour to the air-pump, but probably a better vacuum would be secured than is secured with the ordinary surface condenser and at less expense in circulating water.

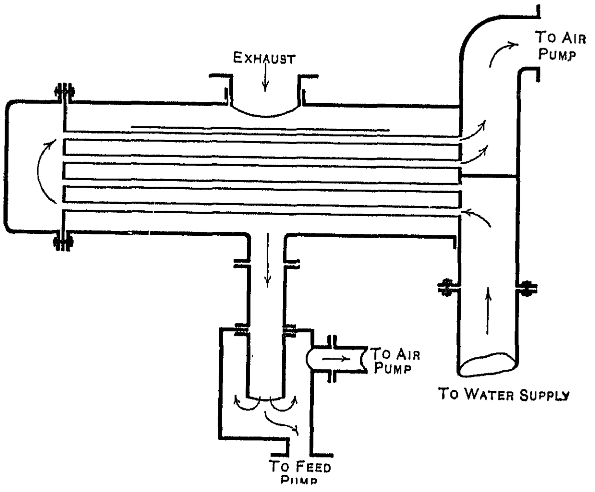


Fig. 1367

The latest form of condenser is the counterflow concentric condenser known as Bracket's. In this apparatus a series of alternately plain and corrugated thin brass tubes are held between two head castings, which are formed with water and steam passages so devised that water and steam find their way along the narrow annular passages between the tubes in opposite directions. Thus each ring column of water flows between walls heated by steam. The efficiency is a maximum, because of the counter-current system, and because the streams of both steam and water are broken up by the one corrugated wall of each passage-way. Several tubes of gradually reducing diameter are placed one inside the other in each nest, and half a dozen such nests of concentric pipes are grouped round one central nest. By this means an enormous condensing capacity may be contained in a few cubic feet of space. The multitude of tubes of various sizes cannot possibly be put together wrongly, and the condenser can readily be dismantled for cleaning.

Mr. Borrowman considers that surface condensers lose much of their efficiency when the tubes are exposed to the flood of water which descends from above. On this account he divides his horizontal tubular condenser into some three or four compartments by means of diaphragms, which each serve to collect the water sent down from the tubes above them, and the water thus collected is delivered by suitable drains to the air-pump without again encountering further steam. This is undoubtedly an improvement, and is a step in that division of duty much on parallel with the principle of stage heating and compound working by which a maximum effect is sought at each stage instead of a general compromise of effects that must be lower than the maximum.

CHAPTER II

AIR-PUMPS

General.—The duty of the air-pump is, primarily, to remove the air which alone prevents a condenser from producing a continuous vacuum. As we have already seen, the air, by which of course is meant all the non-condensable gases, is derived partly from the feed, partly from the injection water, and partly from leakage, and the last source is the principal one in all cases, though with jet condensers and polluted water there may be a considerable volume of carbonic acid and of hydrogen evolved as the products of putrefaction. With surface condensation there should theoretically be very little air to deal with, for the feed to the boiler, once deprived of its air, will not absorb much when hot and continuously circulating.

The action of an air-pump must be considered from the point of view of the expansion of gases by heat and by reduction of pressure.

Let it be assumed that an air-pump has a volume one-half that of the condenser. During the stroke of the pump the air in the condenser would then come to occupy a double volume and would exert one-half its former pressure. Each stroke of the pump would abstract half the air left in the condenser. If it be allowed that the vacuum reads 2 lbs., corresponding to a temperature of 126° F., then, without further air admission, the first following stroke of the air-pump will reduce the pressure to 1 lb., corresponding with a temperature of 102° F. A second stroke would reduce the pressure to $\frac{1}{2}$ lb., which corresponds with a temperature of 80° F. Now if the water in the condenser has a temperature of 100° F., it will be impossible at once to secure a vacuum better than 0.94 lb., because the water will flash into vapour of that pressure and this would be pumped away; but were the process continued, the water would be cooled by this evaporation in time and a perfect vacuum would then be produced. With too big a pump something like the above takes place. There is not sufficient air to fill the air-pump, and its place is filled by vapour. Obviously the vacuum can never be better than that indicated by the temperature, and, in practice, it rarely attains that excellence owing to air leakages. It must be remembered that each cubic inch of air that gets inside any part of a steam boiler or engine will occupy from 5 to 25 times that volume in the condenser, and under the best conditions any further improvement of the vacuum soon overtakes the capacity of the pump further to deal with it. The condenser capacity does not, *per se*, affect the action of the air-pump, but as this latter is intermittent in its action the condenser must be at least large enough to receive all the air delivered to it by the cylinder without seriously affecting its vacuum. Thus with a condenser of a capacity of 1000 the admission of air occupying a space of 100 at the condenser pressure would raise that pressure 10 per cent. If that pressure were 2 lbs. it would become 2.2, showing a loss of 0.2 lb. of vacuum.

Hence the greater steadiness of the vacuum when multiple air-pumps are employed and condensers are large.

The efficiency of an air-pump is greatly affected by the perfection with which it discharges the air it receives. In a plain vertical pump, as fig. 1368, the mixed air and water pass through the bucket and the water settles upon its upper surface, and the air and surplus water are forced through the delivery-valves. When the bucket descends, the delivery-valves on their part are sealed by the water upon them and a very perfect vacuum may be formed above the bucket. But air from below balances the pressures on the two sides of the bucket, passing through the bucket valves to do so; and finally, the bucket descends upon the water collected below it, and some of this is abstracted and forced with the air through the delivery-valve at the next up-stroke. This action, more or less modified according to the style of pump, is common to all air-pumps.

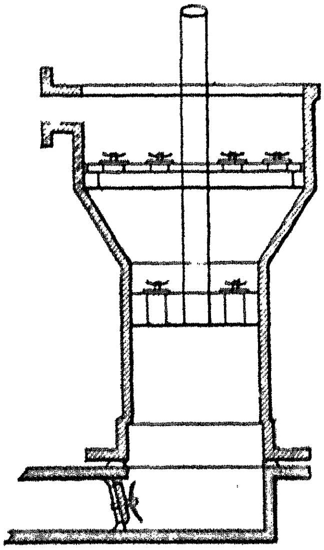


Fig. 1368

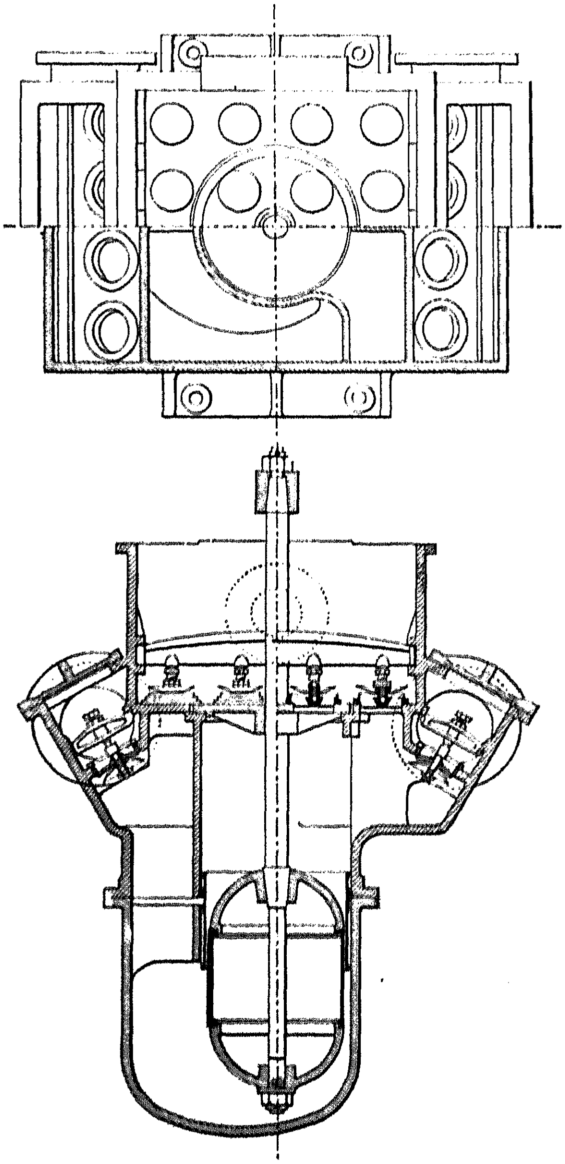


Fig. 1369.—Hick, Hargreaves, & Co.'s Air-Pump

Vertical Air-Pumps.—The plain vertical air-pump (fig. 1368) consists of a barrel with a closed top. This is fitted, in the modern type, with india-rubber valves opening upwards, and exactly similar to the valves of the bucket, which also open upwards. A foot-valve, which opens also towards the air-pump, admits water from the condenser. The action has already been explained, and the principles of all air-pumps are the same.

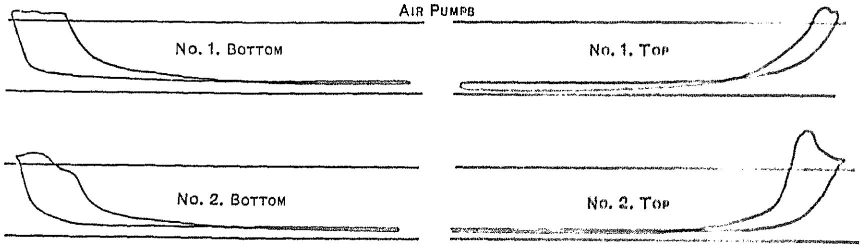


Fig. 1370.—Air-Pump Diagrams

In the displacement-pump of Hick, Hargreaves, & Co., of Bolton, the pump-bucket is an ogival-headed piece, as shown in fig. 1369, working always in solid water in a central sleeve. It acts by varying the volume of the pump casing, into and from which the valves admit and discharge air and water. The valves are always drowned and therefore tight, and the air cannot remain in the pump to destroy its efficiency, but must at once escape through the discharge-valve. This pump was employed in the 3000-horse-power engines made for Messrs. Sassoon of Bombay. The circulating-pump was of the same general construction. The indicator diagrams from the air-pumps are shown in fig. 1370, and prove that the action is good.

Air-pumps of the common type shown in fig. 1368 are not, of course, suitable for high speeds, for which the displacement or plunger type must be employed.

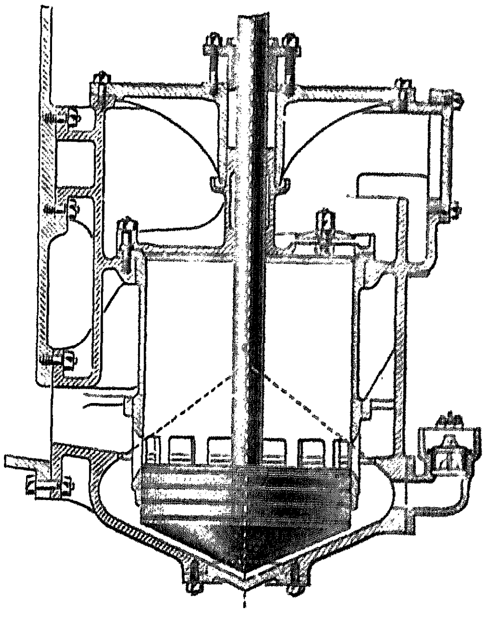


Fig. 1371.—Edwards Air-Pump

The Edwards Air-Pump.—In this pump (fig. 1371) the bucket is solid and has no valves. As it descends, the vacuum, formed above the bucket, is as perfect as the temperature will allow. The buckets are run at a high speed, and when they reach the bottom of their travel they strike forcibly with their pointed conical bases upon the water collected there from the condenser, at the same instant as the bucket uncovers the inlet openings round the barrel. The peculiar curved shape of the lower water passages compels the water, suddenly displaced by the bucket, to rush violently round the passages and into the pump barrel with the air, which also rushes in to equalize the vacuum in the pump and in the condenser. At once, as the bucket rises, the water is trapped above it, and, before the splash is over, mixed air and water are already being ejected at the top discharge. The action is well shown in fig. 1373, which is from a glass model made to show the effect, and in fig. 1372. These pumps work very silently, and are often placed three in line, as in fig. 1374, which shows a triple pump motor-driven. The writer has driven them also by Brown's reduction gear, and they may be belt- or chain-driven. The following figures are given, as they offer useful information as to the speeds possible, and show also the more moderate speeds where these pumps are set to deal with the larger amounts of water from jet condensers. They are from actual tests.

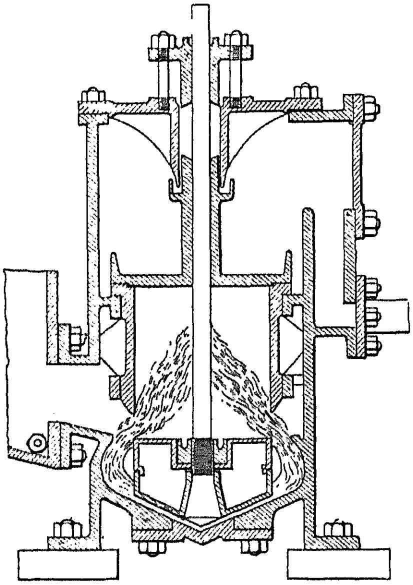


Fig. 1372

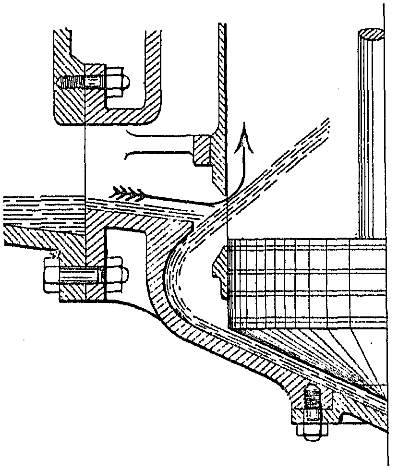


Fig. 1373.—Diagram showing Action of Bucket on Down-Stroke of Edwards Air-Pump

Type of Condenser.	Revolutions per Minute.	Vacuum in Inches.	Barometer in Inches.	Temperature of Air-Pump Discharge in Degrees Fahrenheit.	Pressure due to Temperature in Inches.	Difference between Vacuum obtained and highest Vacuum theoretically possible in Inches.
No. 1 Surface	240	30.2	30.85	65	.619	.03
	250	29.6	30.8	83	1.1	.1
No. 2 Surface	375	28	30.45	107	2.369	.081
No. 3 Jet ...	128	28.25	30	88	1.328	.422
	128	28.375	30	84	1.169	.456

The pump of fig. 1374 has three barrels of 14 inches diameter and 12 inches stroke, and it is run at 150 revolutions per minute, at which speed it is rated as good for a steam supply of 45,000 lbs. per hour from a surface condenser, a figure that points to the provision of 1 cubic foot of air-pump volume for each 1½ lb. of steam condensed. By air-pump volume, of course, is here meant volume generated by the buckets, which, in this case, each generated fully 1 cubic foot per revolution, being single-acting only.

Double-acting Vertical Air-Pump.—The vertical double-acting air-pump of Davy, Paxman, & Co., fig. 1375, has a barrel with central openings, past which the bucket travels on both its up and down strokes. The movement of the bucket away from its barrel-ends creates a vacuum as perfect as the temperature will permit, and the air and water, collecting in the passages from the condenser, rush in as soon as the ports are open. This occurs on both strokes, and the pump can be run as high as 250 revolutions per minute for the 14 × 7 inches three-crank pump. The air and water are expelled from the top of the bucket by the valves in the top cover, and from under the bucket by the valves in the bucket itself, into the interior of the bucket, and thence to the discharge by way of the central trunk, which works in a water-seal, extending above the sleeve in which the trunk works, thereby preventing air leakage by way of the outer surface of the trunk. The upper side of the bucket has a slightly larger area than the lower side, by reason of the trunk, and a little over half the pump duty is done by the top side of the bucket. There are thus no foot-valves in this air-pump other than the bucket itself, which forms its own foot-valve in connection with the openings to the barrel which are

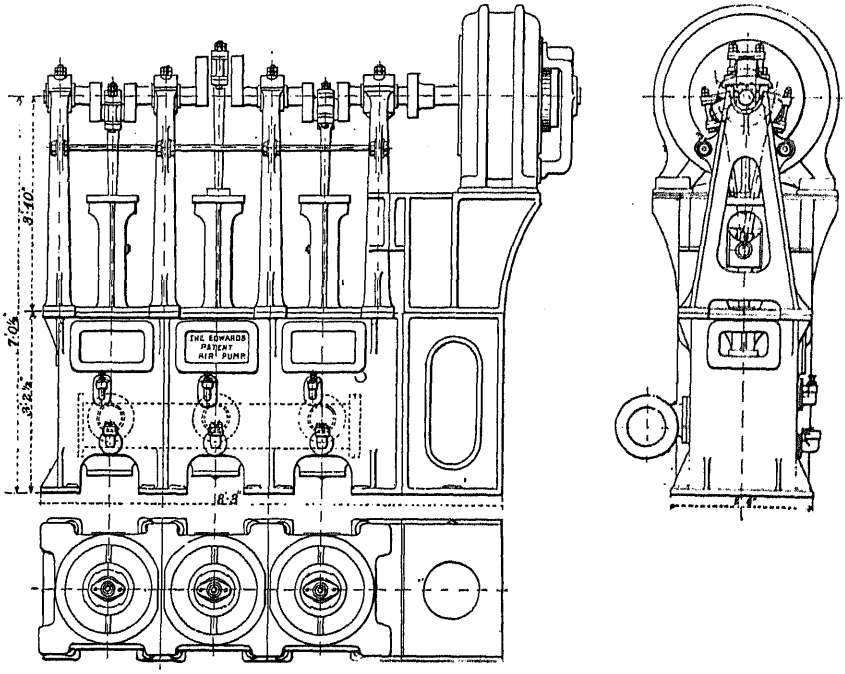


Fig. 1374.—Edwards Triple Air-Pump

uncovered as the bucket moves to and fro. The tendency of modern practice seems to be all in the direction of this device. The pump, being double-acting, may be of only about three-fourths the diameter of a single-acting pump of same stroke to give equal results.

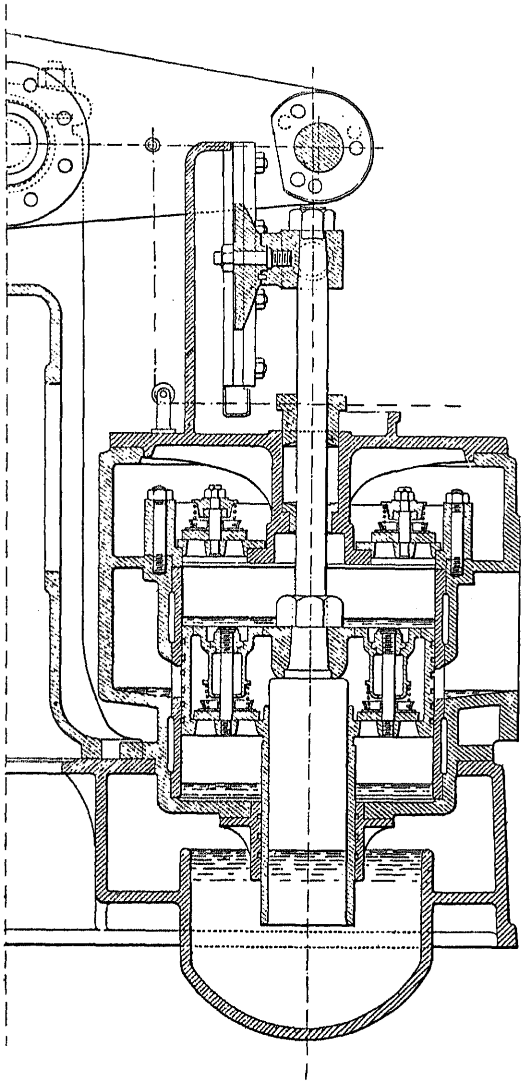


Fig. 1375.—Paxman Air-Pump

Horizontal Air-Pumps.—The generally poorer results given by horizontal air-pumps are more due to bad design than to any bad principle, though it must be confessed, where the horizontal pump is at its best it is so designed as practically to act on the vertical principle when of slow-speed variety.

Thus the pump of Hick, Hargreaves, fig. 1369, could readily be arranged horizontally to act very well.

Mr. M. Longridge, in 1888, pointed out that horizontal air-pumps of the type of fig. 1376 should be so constructed that the velocity of the water in the barrel must not exceed that which is due to a head measured from the centre of the bucket to the upper surface of the water in the end chambers. Tail-rod air-pumps have often a plunger velocity of, say, 600 to 700 feet per minute, and a maximum velocity 1.57 times the mean, or, say, 942 feet per minute in a given case. If the barrel enters squarely into the end chambers this latter velocity is increased by 50 per cent for *vena contracta*, or to 1412 feet, if the water is to follow up the plunger. This points to a head of 8½ feet, which is ordinarily out of the question.

There must therefore be conoidal mouthpieces to the barrels, and too high a velocity of plunger must be avoided. It is obvious that the surface of the water, as it rises and falls with the plunger movement, forms the real working surface, alternately drawing in and expelling the air and water at the delivery-valves.

Tail-rod air-pumps, with a mean velocity as high as 900 feet per minute, have been run. Such pumps may be on the lines of that in fig. 1360a, with ogival-headed plungers round which the water closes as the plungers move, the maximum travel of any particle of water being a distance equal to the radius of the plunger.

In any case the horizontal air-pump of fig. 1376 type can be run from the tail-rod of an engine with a mean piston speed of 400 feet per minute, even with flat plunger as in fig. 1376, if the barrel-ends are flared out, when the water height is only 18 inches; but ordinarily, pumps of this type must be kept moderate in speed and cannot therefore be applied to long-stroke engines. They may be driven by uneven L levers from vertical engines, and they are a suitable form of pump for direct driving by a small steam cylinder in an independent condensing set, for the bucket is always drowned and the action is equivalent to that of a vertical pump.

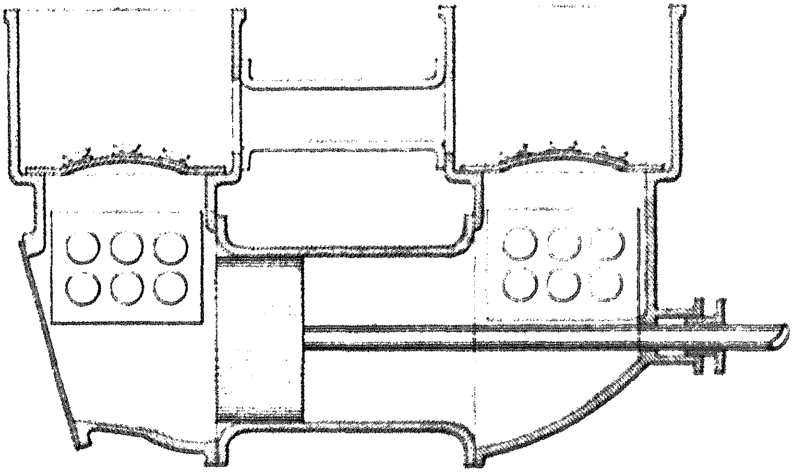


Fig. 1376.—Horizontal Air-Pump.

The horizontal air-pump of Pollitt and Wigzell acts by creating a vacuum behind the solid bucket, and the air and water from the surface condenser above and around rush in through the holes pierced through the barrel, the front end of which is open to the condenser space.

This long-stroke pump is driven direct off the tail-rod of the engine, and has run to as high a speed as 150 revolutions per minute and 880 feet travel. The surface condenser shown in the illustration of this pump, fig. 1378, is notable for two things. In the first place, by means of the peculiar sloping inlet the entering stream is well spread over the whole length of the tubes; and by means of the great length of the outlet it is not invited to take a short circuit to the air-pump, and full use is made of the cooling surface of the tubes.

The circulating-pump attached is driven off a second tail-rod, and is an ordinary double-acting pump. The circulating water should not be drawn from a point so far below the pump that the balance of the atmospheric pressure remaining cannot push the water after the bucket as fast as the bucket travels. The maximum bucket speed is, of course, about half as much again as the mean speed, and may at times be, say, 1300 feet per minute or 22 feet per second. Thus, as $V = 8\sqrt{H}$, we have $\sqrt{H} = \frac{22}{8}$, or, say, 3, whence $H = 9$ feet, which is the head necessary to give a velocity of 22 feet per second. If a pump will lift 18 or even 24 feet with satisfaction, it should not be made in this case to lift more than 9 to 15 feet. A height of 12 feet is specified by the makers of this air-pump as the limit of height from the engine centre to the source of injection water flowing to the jet condenser, when this form is used. These pumps, fig. 1377, run without noise, and are good examples of the possibility of designing air-pumps to run at high speed directly from engine tail-rods.

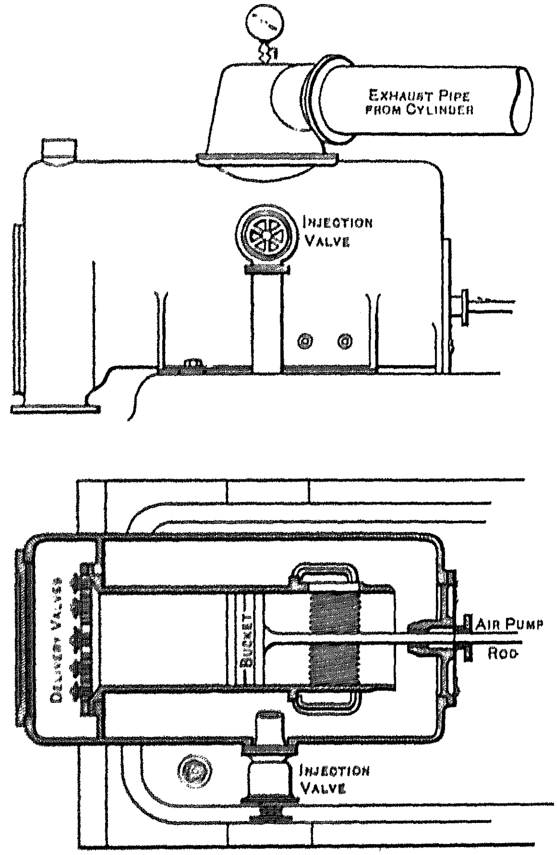


Fig. 1377.—Pollitt and Wigzell's Air-Pump

In the jet-condenser pump of this form the water is not required to follow the bucket as in the style of fig. 1376, but merely enters by way of the diagonal slots under the difference of pressure in the condenser and in the pump-barrel. The speed of the bucket is so great as to spread the water over its surface on the return or discharge stroke, thus pushing out the air first through the numerous small delivery-valves, the water following. It will be here perceived that if a horizontal air-pump of this undrowned type had a slow-bucket speed the water would fall to the lower side of the barrel, and some air would remain in the barrel with the residual water and would vitiate the perfect vacuum on the forward stroke. This explains why ordinary horizontal air-pumps of slow-speed type fall so much in efficiency below the plain vertical pump. Where the bucket moves quickly the water is compelled to spread itself over the bucket surface, and the action of the pump is the same as that of a vertical pump. This pump also does not work in a mass of solid water like the type of fig. 1376, and should not be confounded with it.

In fig. 1379 is shown the complete combination of a compound direct-

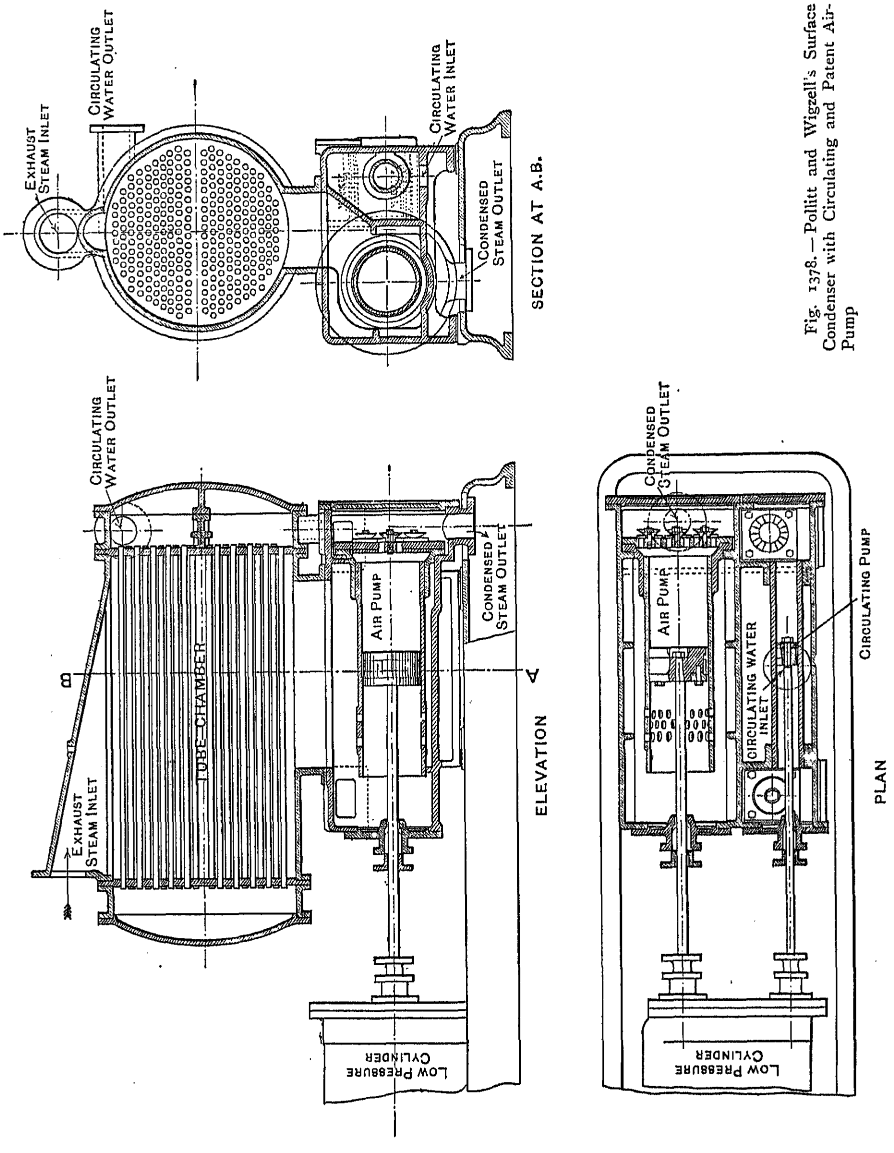


Fig. 1378.—Pollitt and Wigzell's Surface Condenser with Circulating and Patent Air-Pump

acting steam-engine with air and circulating pumps by the Worthington Pump Co. of London. The steam cylinders are 6 and 10 inches diameter respectively, with 10¼ inches air and circulating pumps with 10 inches stroke. The whole is on the duplex arrangement fixed below a surface condenser of 1075 square feet area. Details of the flanged condenser tubes are shown above the condenser.

The Atmospheric Valve.—Modern power plants are usually provided with an exhaust-valve opening automatically to the atmosphere should the condenser fail to act.

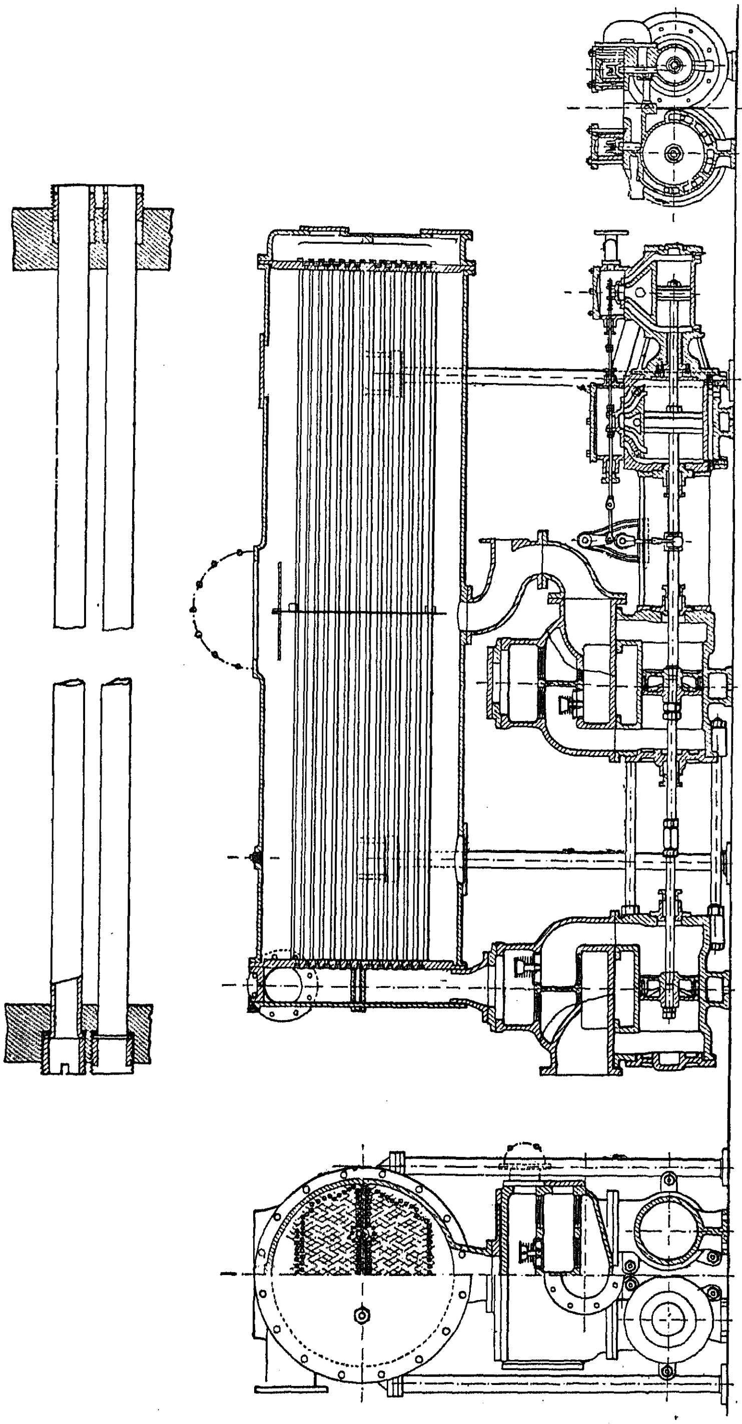


Fig. 1379.—Worthington Condensing Set, complete

Such a valve, by T. Ledward & Co., is shown in fig. 1380. It is held open by a weight, and should have an oil dash-pot (not shown here) to steady it and prevent hammering on its seat. A water-seal ought to be provided to prevent air leakage, also a drain-pipe to keep the water-seal from exceeding an inch or two in depth, or the valve may be so heavily loaded as not to open freely. A glass gauge should show the presence of the water-seal. An excellent atmospheric valve is also made by Thos. Walker of Tewkesbury.

Bucket-Rings.—The vexed question of bucket-rings is one that admits of considerable argument on both sides. In vertical pumps, which always carry some water on the top of the bucket, rings seem to be unnecessary, and the bucket should be deep and grooved with a number of rectangular grooves, which are quite sufficient to prevent leakage, but a horizontal undrowned bucket run at a slow speed might perhaps have rings under certain circumstances, though it is not easy to see the need of rings in most cases. A bucket turned to a nice fit in a barrel, and grooved with rectangular grooves ¼ inch wide, spaced ⅝ inch centre to centre, will leak an inappreciable amount of water. The writer has for years made deep well-pumps in this way, the bucket length being not less than its diameter. Such buckets work with less friction than ring-buckets, and are safe from the break-downs which occur through rings breaking and jamming the bucket.

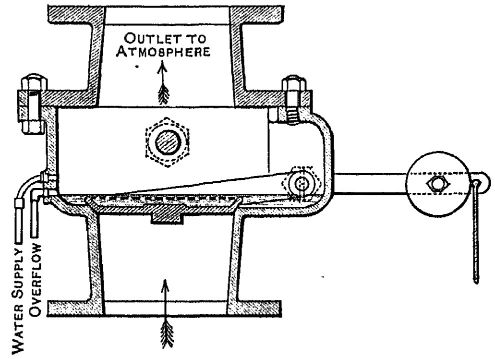


Fig. 1380.—Atmospheric Valve

Circulating-Pumps.—Probably the best circulating-pump, as a rule, in an electric station is the centrifugal pump. Acting, as this does, by the centrifugal action of the water flowing through it, the pump must run at a high speed, and is thus very suitable for direct connection to an electric motor. They occupy little room, and discharge large volumes of water. They are rated by the diameter in inches of their inlet and discharge pipes. A 5-inch pump will discharge 30,000 gallons per hour, and other sizes in proportion, that is to say, in proportion to the square of the size. A general formula for capacity is $G = 3 D^2$, where G is the output in cubic feet per minute, and D is the size in inches. A centrifugal pump will lift water from a depth of 30 feet if cold, but it must be filled with water first, and there should be a foot-valve on the bottom of the suction-pipe. When necessary to force water to a considerable height, centrifugal pumps may be placed in series with each other. The efficiency of a centrifugal pump and electric motor may be approximated at 40 per cent over all when the pump is working freely without throttling of the supply. Centrifugal pumps may, when there is no free fall of water available, be employed to supply water direct to ejector condensers, as described under this head. For very high lifts, which may sometimes be necessary, centrifugal pumps are run in series, and may be made to lift water 1000 feet high.

The Displacement Circulating-Pump of Hick, Hargreaves, of Bolton (fig. 1381), is constructed on the same general form as their air-pump (fig. 1369). Generally a circulating-pump may be of any good design that is convenient to employ, but the centrifugal pump offers itself as particularly convenient for electric driving owing to its high rotative speed and compactness. It cannot be used as an air-pump, but could be employed to drain jet condensers, where a dry air-pump was also used, so long as there was an automatic arrangement, as with the jet condenser of fig. 1364, that would always ensure the full supply of water that the pump could deal with, otherwise the pump might drain the condenser dry and draw only air or vapour, with which it cannot deal.

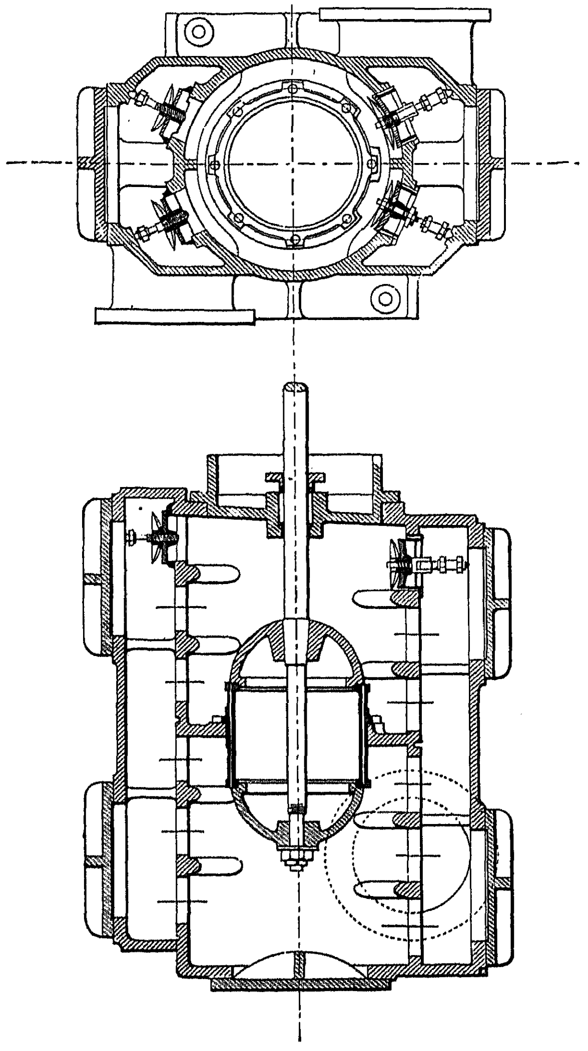


Fig. 1381.—Hick-Hargreaves Circulating-Pump

When employed to raise water to a tank or cooling-tower, the discharge-pipe should taper gradually to about double diameter, so as to diminish the velocity of discharge and economize power by reducing the energy in the outflowing stream.

The Worthington circulating-pump, combined with air-pump, is shown in fig. 1379, already described.

With the advent of steam-turbines it has been thought desirable to secure vacua of greater intensity, and this has led to the employment of compound or two-stage air-pumps. At the Manhattan power station, in

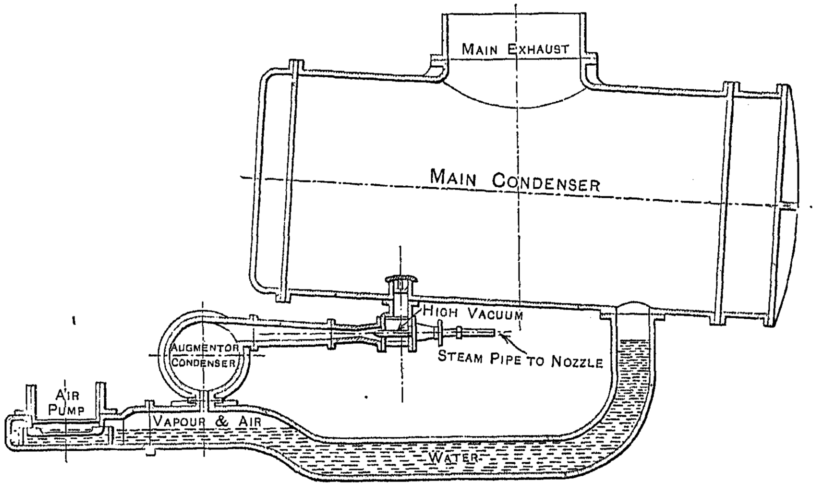


Fig. 1381 A

New York, a rotative dry air-pump of the type of Beale's gas exhauster draws air from the head of the barometric condenser and delivers it to the barometric discharge-pipe in small bubbles which are carried away by the descending column of water. The rotating air-pump does not deliver into the atmospheric pressure but to a point where pressure is only about a third of an atmosphere. With the Parsons turbine a steam jet vacuum augmentor has been much employed, as shown in fig. 1381 A. The condenser is slightly inclined so as to facilitate the fall of the condensed water to the lower outlet, which is formed into a sunk trap as shown. Near the upper end of the main condenser an air outlet is provided, and the air is propelled by means of a small steam ejector through a special supplementary auxiliary condenser and delivered to the air-pump beyond the sunk water-trap. Thus the air-pump gets hold of air already somewhat increased in density above the condenser pressure, and its efficiency and capacity is thus improved. Thus, with a condenser vacuum of 27½ to 28 inches the vacuum at the air-pump may be only 26 inches, the jet having improved the acting vacuum in the condenser by 0.736 to 0.922 lb. per square inch. The tube surface of the auxiliary condenser is one-twentieth that of the main condenser, and the steam consumption is said to be 1½ per cent of the main consumption, while the vacuum is improved ¾ to 1 inch, says Mr. Parsons, or equivalent to 4 to 5 per cent of the main power. The circulating water is made fifty times the weight of the feed. With reciprocating engines an equal improvement of vacuum would not represent a greater gain than 3 per cent gross or 1½ per cent net, even with so low a mean pressure as 30 lbs. referred to the low-pressure cylinder.

A very small addition to the pressure of air delivered to an air-pump will add enormously to the capacity of the pump. Thus, if a final ½ lb. absolute pressure remains in the main condenser, and this is built up by a jet or a rotatory exhauster to 1 or 1½ lb., the air-pump will take in double and treble as much air each stroke.

CHAPTER III

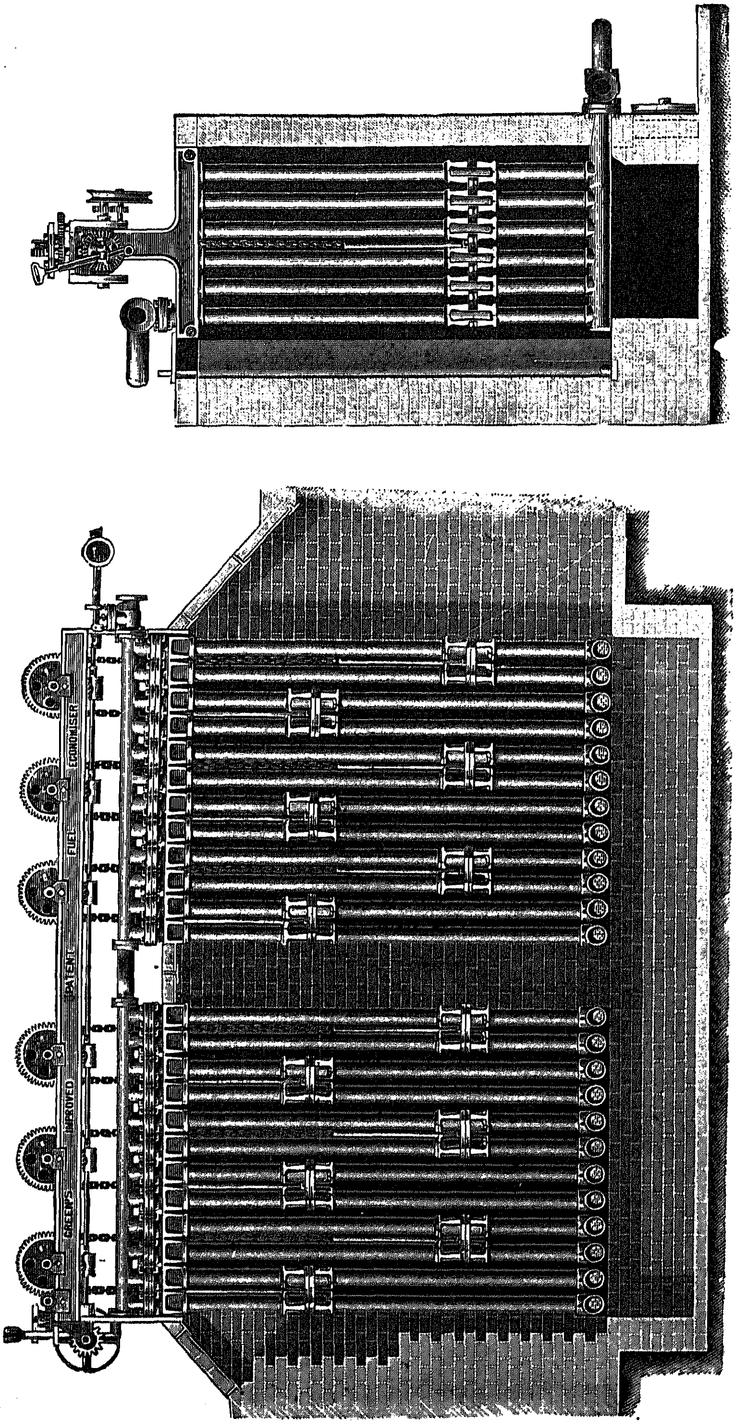
ECONOMIZERS AND FEED-HEATERS

Though the term economizer has become almost confined to the flue feed-heater which economizes otherwise waste heat, it may properly be applied to any apparatus that saves something that would otherwise be wasted. For non-condensing engines the exhaust steam is led through a vessel which contains tubes through which the feed-water is pumped, and in this manner the feed may easily be heated to 212°, and considerable economy secured.

Where condensing engines are employed, the exhaust steam is sometimes made to pass through a small surface condenser on its way to the main condenser, and the feed-water is passed through this and may thus be given a preliminary heating on its way to other feed-heaters. Such preliminary heaters are often combined in the same piece as the surface condenser.

Green's Economizer (figs. 1382, 1383).—The economizer, as now commonly understood, consists of a series of cast-iron pipes secured at each end into cast-iron boxes. Such a set of pipes, to the number of four, six, eight, ten, or twelve pipes in width, constitutes a section. The boxes are flanged at each end, and any number of sections are joined together by a single inlet-pipe communicating with each bottom box, and an outlet-pipe which carries off the water from the top boxes. Thus all the pipes are in parallel, and water flows through them, probably in accordance with the ability of each pipe to add heat, those pipes in the hottest part of the flue doing the most work. The number of sections which are put together to form one apparatus is always a multiple of four. Thus in an economizer built up of sections having eight pipes, the total number of pipes must be 32, or 64, or 96, and so on, always increasing by four sections, the reason for which is that there are soot-scrapers on each pipe, carried by bars slung by chains. The chains hang between the sections, and each chain carries a transverse bar, on which the scrapers are hung. The chains and their load are balanced by the chains and scrapers of the next pair of sections, being hung over a wheel above. To each wheel there are thus hung the scrapers of four sections, and the wheels are slowly revolved by worm-gear driven from a shaft, which is automatically reversed by a clutch and lever, so as to keep the scrapers in constant up-and-down movement.

On the outlet-pipe is placed a safety-valve, and a cup for mercury to hold the outlet thermometer. On the lower or inlet-pipe is placed a blow-out valve to blow off mud, and also a thermometer pocket. Provision for expansion is made about every twelve sections by a horizontal U bend in the junction-pipes, and in the more extensive economizers there are two or even three separate apparatus, which are connected in series. This ensures that the waste gases shall be better utilized, and yet does not put any serious obstruction to the flow of water.



Cross Section

Sectional Elevation

Fig. 1382.—Economizer or Flue Feed-Heater

Each pipe is $4\frac{9}{16}$ inches external diameter, and 9 feet long between the sockets of the top and bottom boxes. An economizer has 10 square feet of surface per pipe, and weighs 280 lbs. per pipe, and holds 6¼ gallons per pipe. It is usual practice to provide 96 to 120 pipes for each 7 to 8 feet diameter Lancashire boiler. Another rule is to provide four pipes for each ton of coal burned per factory week. Otherwise their surface may about equal the heating surface of the boilers.

When the feed-water is supplied to an economizer at a less initial temperature than 90° to 100° F. the lower ends of the pipes and the bottom boxes act as condensers to the steam in the flue gases, and rapid and severe corrosion is caused which soon destroys the whole economizer. To obviate this a small pipe is brought from the outlet end of the economizer to the feed-pump suction, so that a part of the heated water is returned to the pump to raise the inlet temperature of the feed-water to the economizer to at least 90°, or more if found necessary.

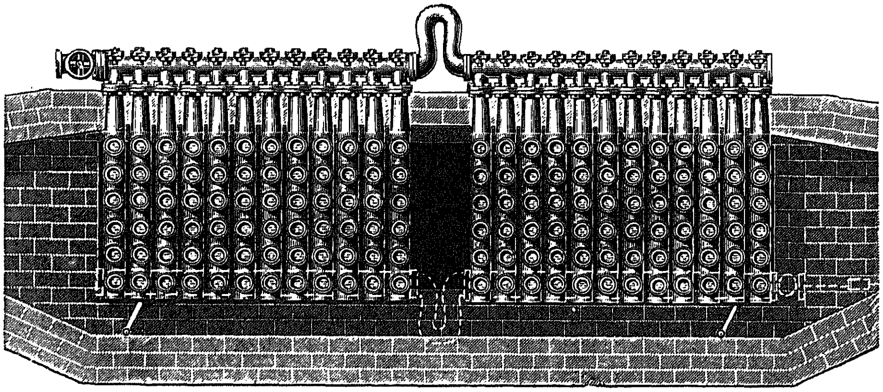


Fig. 1383.—Plan of Economizer or Flue Feed-Heater

This by-pass pipe is not required where the feed is drawn from a surface condenser, or even from the hot-well of a jet condenser air-pump or other feed-heater.

Where a chimney draught is employed the gases should not be cooled below 350° F., but where a fan-induced draught is used the waste gases are best cooled to as low a point as possible, and further economizer tubes may be employed if thought worth while. In ordinary practice it is found that the flue gases are cooled 250° to 300° F., and the feed-water is heated 150 to 220 F. degrees. The economy due to feed-heating is measured by the ratio of added heat to the total heat in the steam. Thus, if water is available at ordinary temperatures it will have a total heat of about 1100 B.T.U. above ordinary temperature of a pond or river.

The addition of 140° F. to the feed-water in its passage through the economizer shows an economy, therefore, of one-eighth or 12½ per cent, and usually 15 to 20 per cent economy is to be obtained in a well-arranged and well-kept economizer.

In passing through the economizer the velocity of flow of water is very low—about ¼ inch per minute is common. The scale-forming salts are driven out of solution by the heat, and in time the pipes become thickly encrusted and may be bored out. Makers of economizers undertake this work by means of multiple-spindle boring-machines, which bore out a section of pipes at once.

In setting an economizer there should be provided below it a soot or dust chamber, if possible 2½ to 3 feet deep. On one side at least there should be a passage admitting an inspector along the full length of the apparatus. This passage is closed by narrow dampers at each end. To enable the economizer to be laid off for cleaning or repairs, there is a by-pass flue direct to the chimney. This may, if needful, be carried below the economizer. Where the furnace arrangements are faulty the economizer becomes very sooty, and sometimes it becomes coated with a tenacious coating of a pitchy quality. Economizers may be burned clean by drying them out and allowing the hot gases to pass through for several hours.

This burns off all dirt, but is an operation to be very carefully performed or the economizer may be injured. Where boilers are so set as not to smoke, it has been found that the scraper-gear and scrapers can be removed, the pipes remaining clean.

The pipes are fixed in their boxes by turned taper ends inserted in bored taper sockets, and a pipe can be cut out if broken, and replaced. In this case the top box joint must be remade in cement.

The following figures may be taken as the performances of economizers at various works:—

No. of Pipes per Boiler.	Gas Temperature.			Water Temperature.			Total Number of Pipes.
	Inlet.	Outlet.	Fall.	Inlet.	Outlet.	Rise.	
32	600	480	120	96	240	144	64
80	580	290	290	98	215	117	320
72	630	520	110	93	217	124	144
75	640	460	180	90	225	135	224
52	570	388	182	107	170	63	208
72	560	360	200	54	204	150	72
Average of a large number of economizers, including those above—	557	381	176	93	202	109	

The maximum recorded rise of feed temperature noted in one year's inspections of economizers by the Manchester Steam Users' Association was 187° F.

Internal Scrapers.—In the economizers of Roberts & Co., of Dukinfield, flat steel scrapers, which fit closely down the whole length of each pipe, are revolved slowly to and fro every few minutes by means of worm-gearing. This it is claimed effectually removes scale as quickly as it is formed, and the scrapings drop down into the bottom boxes and may be thence removed. In this economizer there are two rows of pipes to each top and bottom box. One worm drives two pipe-scrapers, and each worm-shaft is driven by another worm and wheel outside the boxes.

Fig. 1384 shows an economizer in cross-section. Fig. 1385 shows the Roberts internal scrapers and gear.

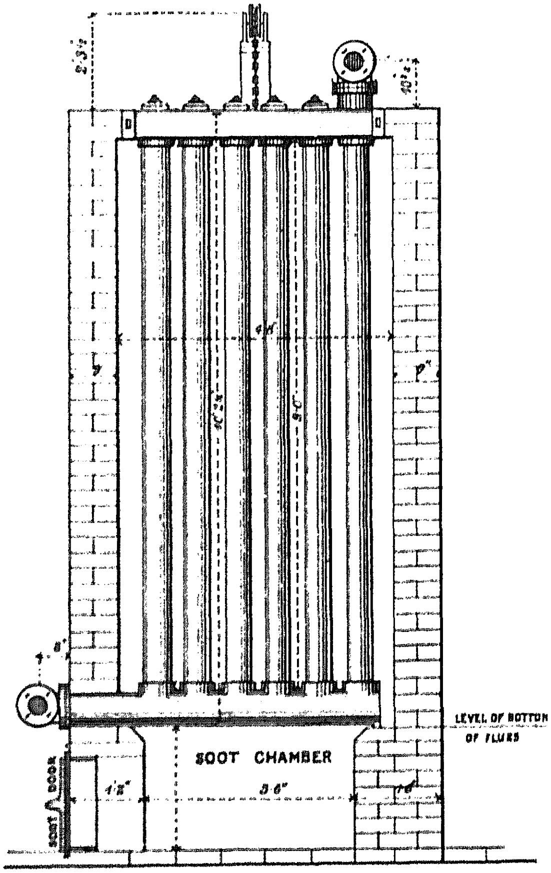


Fig. 1384.—Roberts's Economizer. Section showing brickwork with dimensions

As regards the internal scraping of economizer pipes, the advantage is double. In the first place, the gradual reduction of efficiency which must inevitably take place where incrustation is allowed to occur, is prevented. The economizer always works at full efficiency. Secondly, the time required to bore out an economizer is saved. During this time the apparatus is of course useless, and in a hard-pressed station the cold feed might prevent full load being carried by the station, not to name injury to the boilers. In practice it is not desirable to employ economizers with pipes not circular in section, for they can neither be kept scraped nor can they be bored out; acid alone and scraping can be employed. Unless combustion is so perfect that the outside scrapers can be dispensed with, these ought not to be worked intermittently or they will probably cease to work at all, the chain-wheels slipping under the chains. The economizer shaft is intended to be run at 55 revolutions per minute, and the power required is only small and is best applied by means of an electric motor. At Bournemouth Mr. H. M. Sayers drove his economizers by an electric motor through a worm and wheel. The wheel is best made of phosphor-bronze, and the worm of steel, polished and run in oil. Enclosed motors should be employed, and chain-gearing may be used if preferred to worm-gears, but belts or ropes are unsuitable as a rule by reason of the dust and heat. Frequently the economizer is driven by a small steam-engine, but the motor is much more economical.

Experiment has shown that there is practically no difference in the performance of an economizer when all the pipes are in parallel, and when the various sections are placed in series.[1]

Mr. Longridge found that if the weight of feed per hour be multiplied by the rise in temperature, and the product divided by the square feet of economizer heating surface, the result gave an average transmission per square foot per hour of from 876 to 1095 thermal units. He deduced from this that where the heating surface of the economizer is about equal to that of the boiler, an average transmission of 1000 to 1300 thermal units per square foot per hour may be expected, where about $\frac{3}{4}$ to 1 lb. of coal is burned per square foot of boiler heating surface.

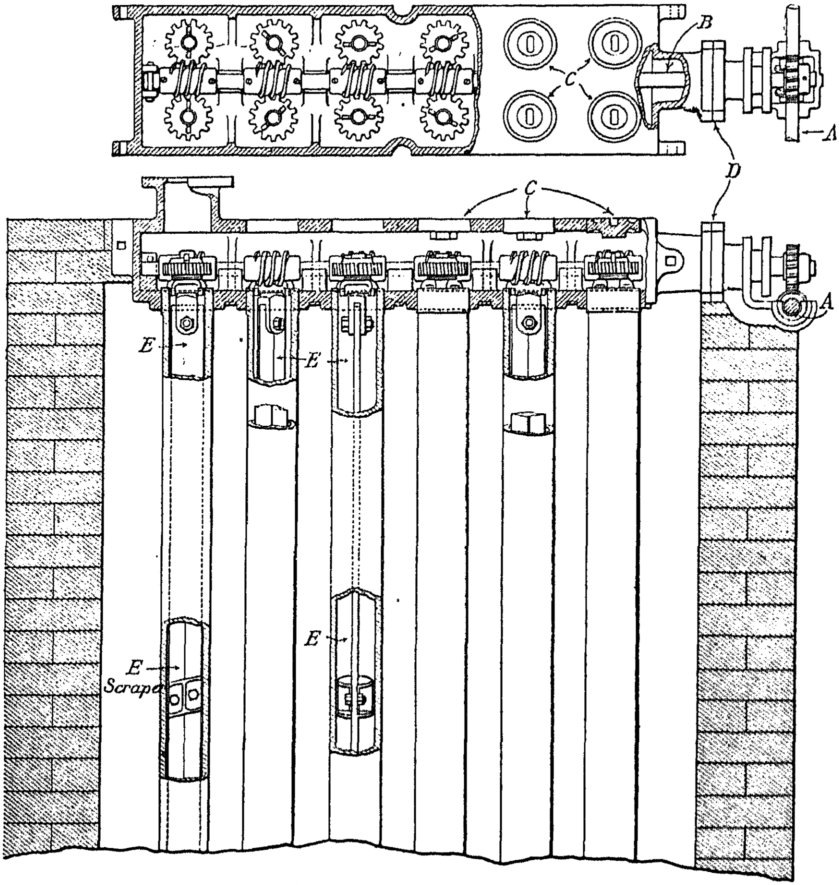


Fig. 1385.—Internal Scrapers to Roberts's Economizer

Fig. 1386 shows an economizer with Lancashire boilers, chimney, and by-flue, with feed-pump and pipes. The expansion bends at the middle are to be noted.

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[1] See Report for 1894 by Mr. M. Longridge of the Engine, Boiler, and Employers' Liability Association.

Feed-Heaters.—When steam-engines are worked non-condensing, and discharge their exhaust at 212° F., each pound of steam contains 967 thermal units which it can give up in condensing to water at 212°. As a temperature of 50° to 100° is common to feed-water it is obvious that the feed can be raised to a temperature of 212° with less than one-fourth of the heat available in the exhaust. The usual manner of doing this is to pass the exhaust steam through a vessel containing pipes through which the feed-water circulates.

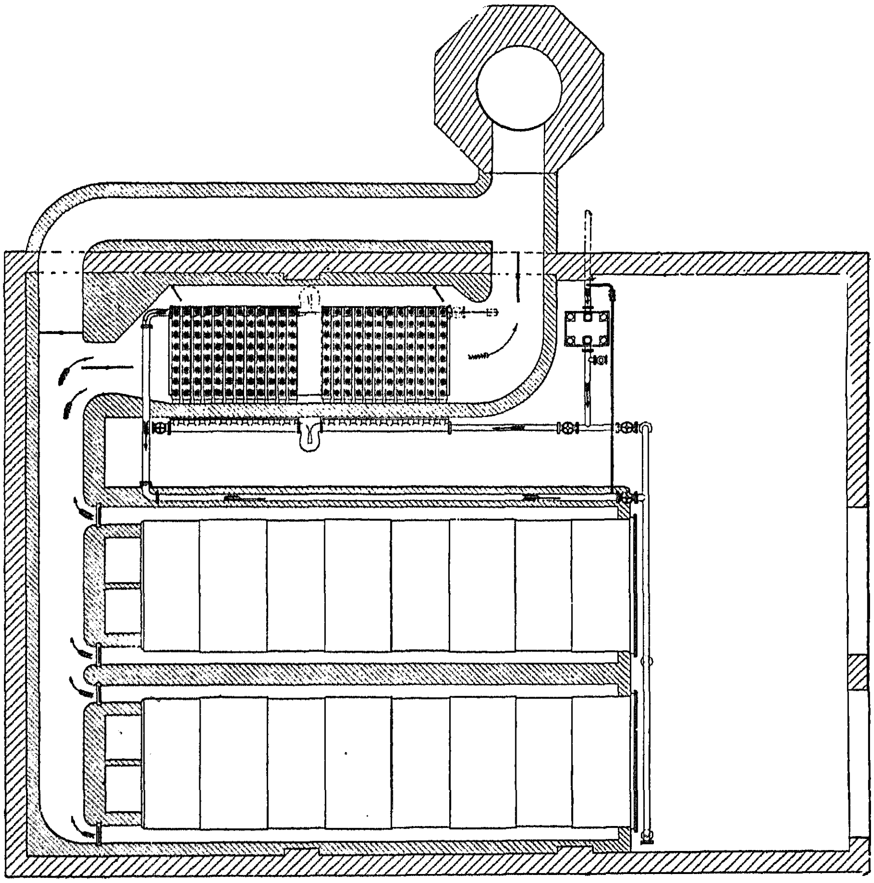


Fig. 1386.—General Plan of Boilers and Economizers

The use of evaporators for make-up of boiler feed has brought forward the subject, and considerable experiment has been made on evaporators and on feed-heaters. Mr. James Weir has advocated and carried out the abstraction of steam from the intermediate receivers of compound engines and the use of the same for feed-heating and evaporation. In this way he has secured the benefit of the work done in the high-pressure cylinder, and he then returns all the heat left in the steam to the boiler. Feed-heating by live steam from the boiler has been found to save from 5 to 10 per cent.

The Pure Water Circulation Economizer.—Having regard to the difficulties attendant on heating feed-water having lime salts in solution it has been proposed to fill economizers with pure water only, and to circulate the same continuously, a counter-current cooler being placed in series with the economizer. Through the cooler is passed the true feed, which being heated away from contact with flame temperature will deposit its scale in soft condition.

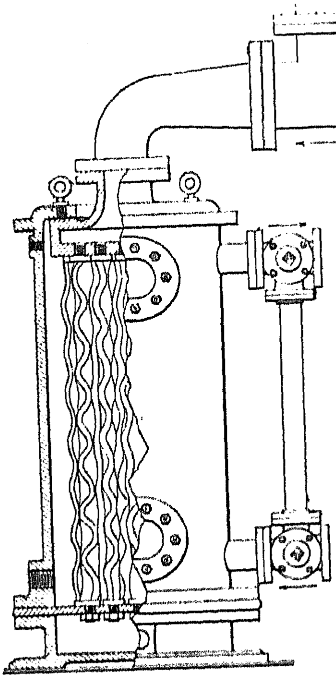
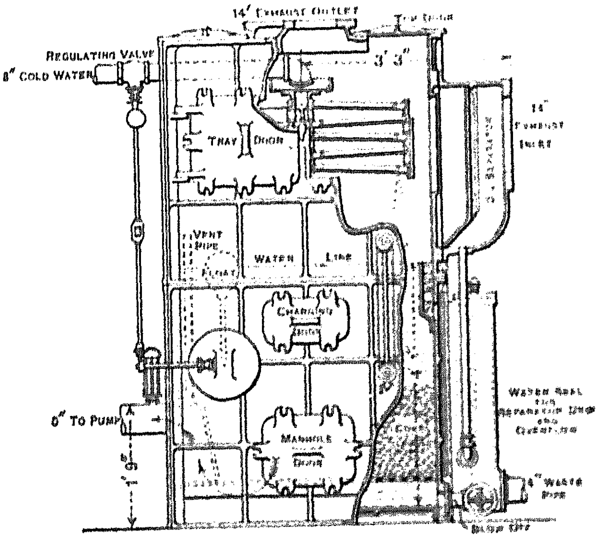
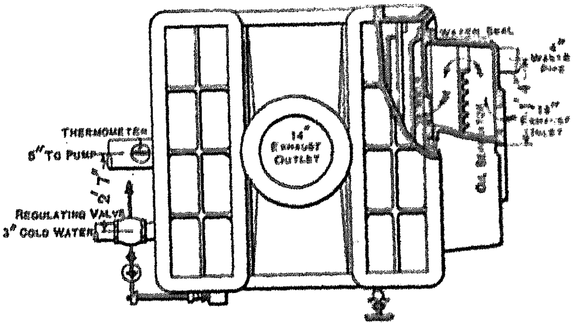


Fig. 1387.—Row Feed-Heater

In such an arrangement the feed-water, if not heated to 212°, should be passed through a further process of live-steam heating in order to compel it to drop its scale-forming matter before it enters the boiler. Where water is not softened chemically, the above system is well worth attention, the heat of the pure circulating water being abstracted by counter-current surface heaters. The economizer circulation being a closed circuit, the pump to maintain circulation would require but small power. Indeed the ordinary circulation due to heating and cooling of the circulating water may be sufficient, though a more rapid movement is conducive to efficiency.



FRONT ELEVATION



PLAN

Fig. 1388.—Cochrane Feed-Heater

The Row Feed-Heater.—In this heater the feed-water is passed through indented tubes, which have a better efficiency than plain tubes.

With steam at 62 lbs. pressure a quantity of 11 gallons of water was raised to boiling-point in 11 minutes by means of plain tube surface, and in 5½ minutes with the Row tube. To evaporate 2½ gallons these required 24 and 11¾ minutes respectively. The indented tube is thus doubly as efficient in transmitting heat. It also has some longitudinal elasticity. Practically the same ratio was found for exhaust steam. The Row feed-heater, made by Royle of Irlam, is shown in fig. 1387. The exhaust steam is sent through the tubes, and the water enters the casing and flows upward against the current of descending steam. The action of the Row indented tube is doubtless to compel more perfect contact with the tube surface of the steam passing through the tubes. Where water is passed through the tubes their peculiar form compels that turbulent

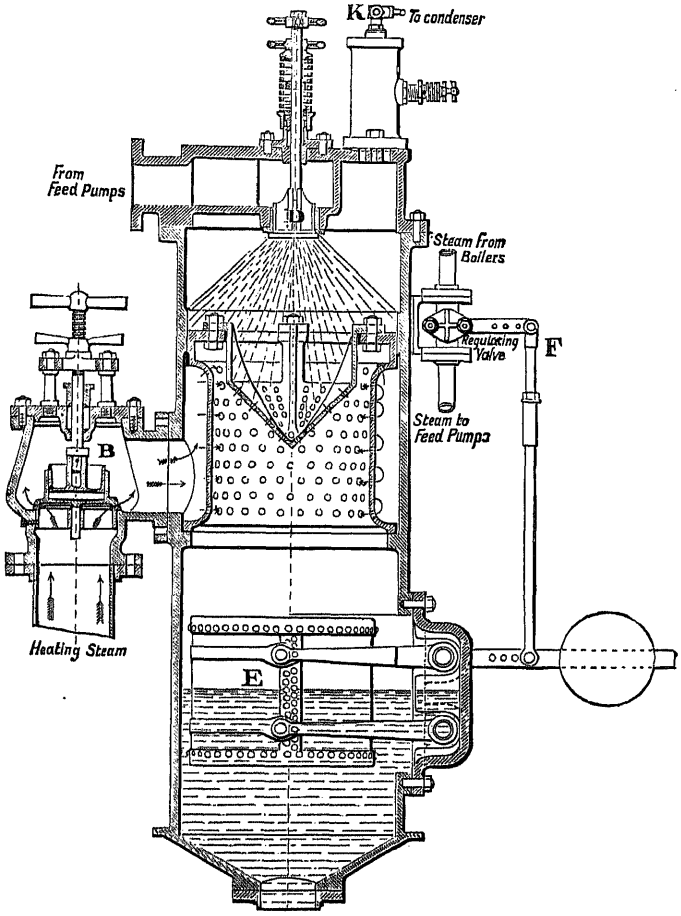


Fig. 1389.—Weir Feed-Heater and Purifier

flow which has already been alluded to as assisting to render small tube condensers more efficient than large tubes by virtue of the turbulent flow more easily induced in small tubes.

The Row tube is applied also to live-steam heaters, surface condensers, cooling-tower, evaporative condensers, and other uses, such as make-up feed evaporators.

The Servé ribbed tube probably acts partly by inducing turbulent flow as well as by increase of heat-absorbing surface.

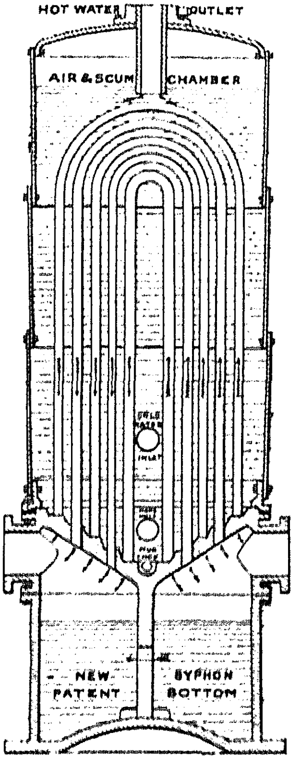


Fig. 1390.—Berryman Heater

Cochrane Feed-Heater is a combination feed-heater and purifier in which exhaust steam is brought directly into contact with the water to be heated. As shown in fig. 1388, the steam is admitted to the casing through an oil separator, and the cold feed is trickled in over a series of trays into actual contact with the steam. Such a system is far more effective than any surface heater can be in a given space. At the base of the heater is a coke filter. Water thus thoroughly heated is deprived of its contained crust-forming salts, which deposit in the trays or can be filtered out below. Heating the feed deprives it also of its occluded gases, which are known to be one, if not the chief, agent in boiler corrosion.

The Weir Feed-Heater (fig. 1389) acts by contact of steam and water. It is self-regulating by means of a float-actuated valve, which acts as the feed-pump draws off the water from the base.

The cold feed enters in a spray, as shown, and makes contact with the feed entering round the perforated casing, air and gas is drawn off above, and this air or gas is corrosive. The steam is drawn from the low-pressure cylinder casing and from auxiliary engines at sea very frequently. In a closed heater the steam has a temperature above 212° F., and may therefore heat the feed above atmospheric boiling temperature. Thus in a particular case the receiver pressure was 16.7 lbs., and the temperature 218°. Water at 100° was taken from the hot-well and raised to 218° in the heater. As the total heat in 1 lb. of receiver steam is 1080 measured from 100°, and the water is given 118° of temperature, this represents 10.94 per cent of the total available heat of the steam. The condensed steam goes back to the boiler with the feed. This method of feed-heating represents an increase of efficiency of the first cylinder of a compound engine, or of the first two cylinders of a triple engine, which have taken work out of the steam that is abstracted from the casing. Where there is no economizer, as at sea, there is no question whatever as to the benefit of feed-heating, and to Mr. Weir is due the attention that has been drawn to the action of the liberated gases in causing corrosion of boiler tubes. Whatever the

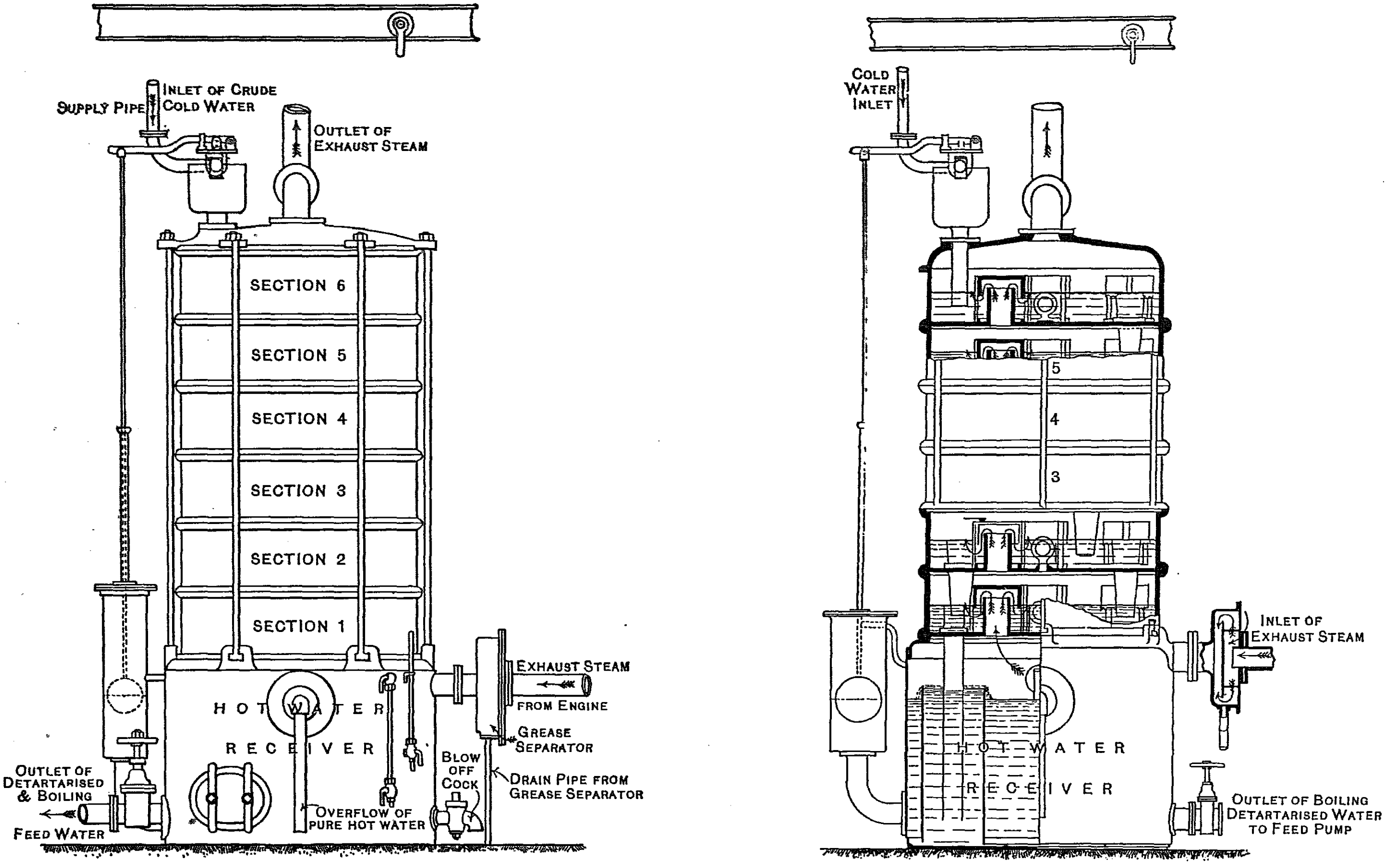


Fig. 1391.—Chevalet-Boby Detartarizer and Heater

cause, the freshly liberated gas appears to have a peculiarly active influence in corroding boiler plates and tubes.

The Berryman Heater.—This form (fig. 1390), made by Wright & Co. of Tipton, is well known. It is a good type of heater, for the tubes are free to expand without stress, and the whole dome cover can be raised and the tubes cleaned of scale very readily.

The Chevalet-Boby Detartarizer and Heater (fig. 1391) is a feed-heater in which live steam is the agent employed, or exhaust steam from non-condensing engines, or from the intermediate receiver of compound engines. The water must be heated to 212° F. to deposit carbonate of lime, and soda is used sufficiently to deposit any lime sulphate. The apparatus consists of a series of trays (fig. 1392) in a vertical cylinder, in which the feed-water is exposed for some time, a cylinder 4 feet diameter and 10 feet high having a capacity of 660 gallons per hour for water up to 18 grains per gallon. The water overflows from each tray to the one lower down, and deposits its load of scale on the overflow spouts and all the interior surfaces of the trays, whence it is easily removed by dismantling the apparatus.

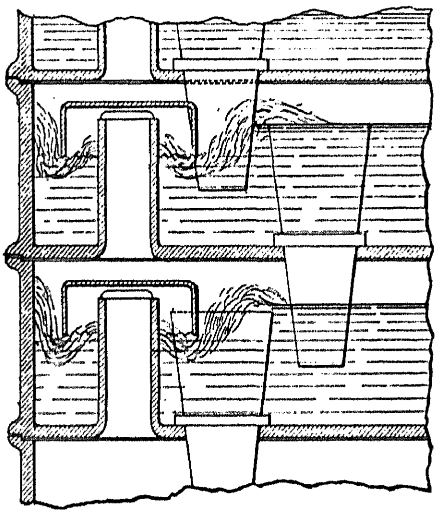


Fig. 1392.—Details of Trays and Tubes

The steam passes upwards through the apparatus, entering each chamber through the vertical spout and under the lip of the inverted pot, and so on to the next tray. The oil introduced with the exhaust steam is absorbed by the scale. This or kindred apparatus would be very suitable to follow the heat exchanger of the pure-water system of economizer circulation, to add the final temperature to the steam, and supply a locus of deposit for the lime salts. This detartarizer is made by Mr. W. Boby of Salisbury House, London, E.C.

Modern practice demands that no water shall enter the water space of a boiler unless it is already heated fully to the boiler temperature. When, therefore, after being first delivered warm to the engine hot-well, the feed is next heated by an economizer, and is still less hot than the boiler, it may then be passed through a vessel into which steam at boiler pressure is admitted, and thereby heated to the full temperature of the boiler. Normand, the celebrated French engineer, first introduced the system of heating the feed to a boiler by the steam from that same boiler, and he determined an economy of 10 to 15 per cent from what may appear as simply paying out of one pocket into another. But there is some peculiar and not understood effect, and undoubtedly a boiler which is only asked to add latent heat to the water will do more and better work than if it is fed with water below its own temperature.

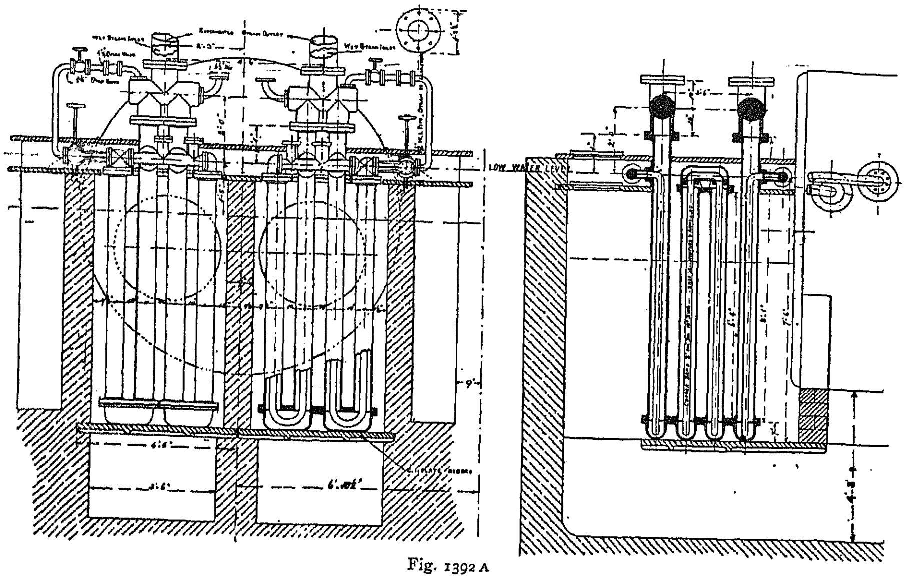


Fig. 1392 A

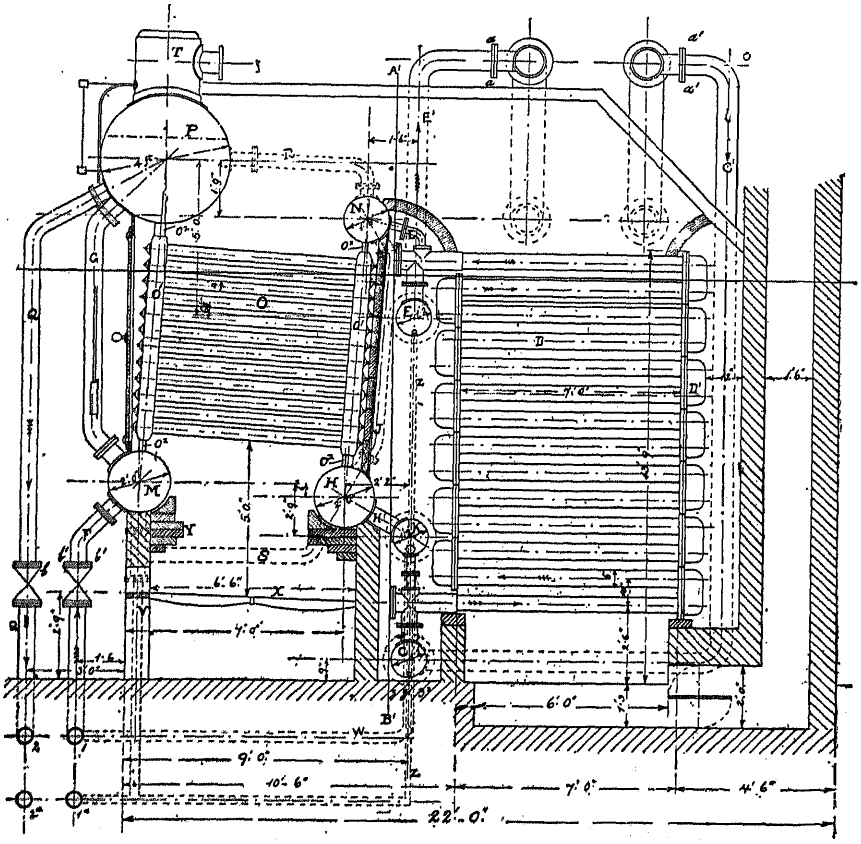


Fig. 1392 B

The final stage in feed-heating may be variously carried out. In the best practice, where controllable superheaters are in use, the feed-water enters the boiler through the water control pipes of the superheater. Thus in fig. 1392 A, which shows a Cruse superheater behind a Lancashire boiler, the internal control pipes are maintained full of circulating water by means

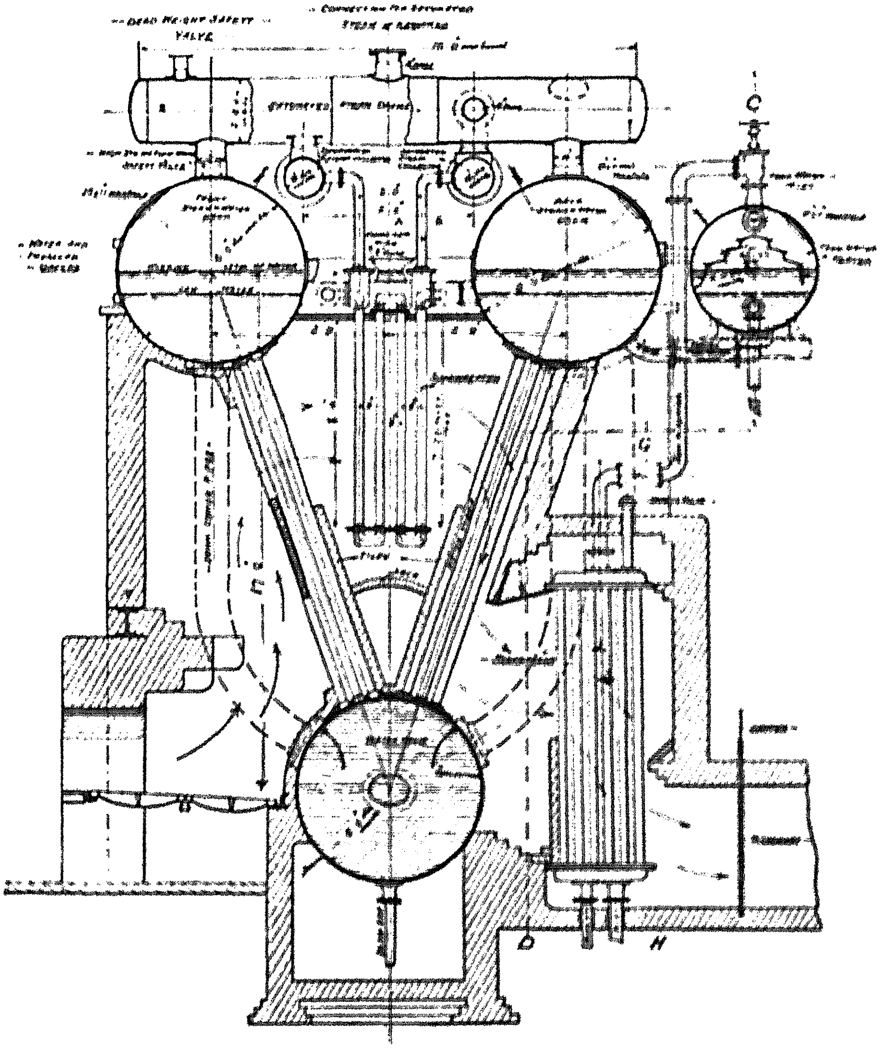


Fig. 1392 C

of a superheated steam propeller or inspirator, which causes water to flow from the boiler through the control pipes and back to the boiler, the feed-water, more or less hot, being pumped into the circulation system *en route.* In the case of separately fired superheaters on the same system, the feed-water to the whole battery of boilers passes into the water foreheater or furnace gas attemperator (fig. 1392 B) and acquires full boiler temperature before it passes on to the boiler; while in a self-contained boiler, superheater, economizer, and feed-heater (fig. 1392 C) the water after it leaves the economizer passes to a drum external to the boiler proper, but in which boiler steam freely enters, and is exposed to the feed-water which is distributed in thin sheets as it descends over sloped plates. The object to be attained in every case is the acquisition by the water of the temperature of the steam in the boiler before entering the water space of the boiler, thus confining the boiler solely to the one duty of evaporation.

In the modern system of stage heating it is of little importance how the water acquires its temperature so long as it finishes at boiler temperature. In marine work, where the economizer is not known, it is very usual to heat the feed-water by means of steam abstracted from the intermediate receiver or low-pressure casing—steam which has already done perhaps half its work in the high-pressure cylinder. And this heating may advantageously precede the final heating, either by boiler steam direct, or by the heat obtained in the control pipes of the water controlled or Cruse superheater.

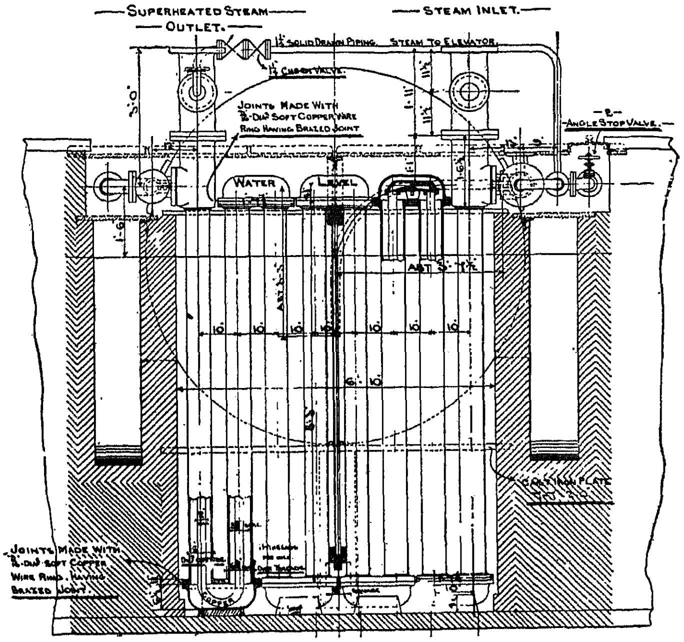

SUPERHEATED STEAM OUTLET.
STEAM INLET.
1½" SOLID DRAWN PIPING.
STEAM TO ELEVATOR.
1½" CHECK VALVE.
JOINTS MADE WITH ⅜" DIA. SOFT COPPER WIRE RING HAVING BRAZED JOINT
2 ANGLE STOP VALVE.
3'-0"
1'-11"
11½
1'-6"
WATER LEVEL
ABT 5'-1½"
10
6 10
COPPER
CAST IRON PLATE
JOINTS MADE WITH ⅜" DIA. SOFT COPPER WIRE RING, HAVING BRAZED JOINT.

Fig. 1392 D

CHAPTER IV

WATER COOLING

It often happens that water sufficient for condensing purposes is not available. In such cases, and where the evaporative condenser is not employed, it becomes necessary to re-use the condensing water and cool it after use. Four methods are commonly employed, viz. the pond, the tower, the sprayer, and the atmospheric.

The first is simply a store of water of sufficient area and depth to enable its contents to lose heat as quickly as they receive it. The inlet from the condenser discharge is placed as far as possible from the outlet to the condenser.

Towers are of two forms, and usually contain extensive surfaces of drain-tiles, wire-mats, or thin boards, over which the water to be cooled is made to pass in thin films. These are contained in a tower which is carried up some height, and serves to create a draught which causes the passage of air through the wetted structure, and this air carries off the heat by absorbing moisture, each pound of which takes up about 1000 heat units in assuming the form of vapour. In other towers air is blown in by fans at the base of the tower. The atmospheric cooler is usually of thin boards unenclosed, and acted on solely by the wind.

Sprayers are small jets or nozzles in which is placed a small bit of spiral sheet metal, which gives a twist to the escaping jet forced through the small nozzle. A large number are employed on one system of pipes. The jets of water split up into fine drops, and cool in falling through the air. The nozzles add considerably to the work of the circulating or air pump, owing to the necessity of forming the jets under pressure.

Whatever the system employed, the principle is that of rendering heat latent by assisting the evaporation of the water to be cooled, as each pound of water which disappears carries away some heat in sensible form and 1000 heat units as the latent heat of evaporation. The annexed table gives figures relative to the moisture-carrying capacity of air.

Temperature	100°	90°	80°	70°	60°	50°	40°	30°	20°	10°
Weight of 1 cubic foot of saturated air in grains	19.84	14.85	10.98	8.01	5.77	4.10	2.86	1.97	1.30	0.84
Percentage of water deposited for fall of 10°	25	26	27	28	29	30	31	34	35	34.5

20,000 cubic feet of saturated air at 90° contains 47 lbs. of water.
At 60° the same volume contains 17 lbs. only.

The weight of vapour that can be carried off by 100 lbs. of dry air is given for various temperatures, as below.

Temperature	32°	41°	50°	59°	68°	77°	86°	95°	104°	113°	122°	131°	140°
Moisture ...	.380	.540	.765	1.05	1.45	1.95	2.69	3.63	4.84	6.45	8.56	11.38	15.14

The actual amount that it is possible to absorb varies with the amount of dryness of the air, and in misty weather a cooler loses much of its efficiency.

For practical purposes the weight of air may be taken on the basis of 13 cubic feet per pound at the ordinary temperature in this country.

In a good tower the escaping air will be nearly as hot as the entering water, or say 110°, and it should carry off 5 per cent of its weight of vapour. Hence each 100 lbs. or 1300 cubic feet of air should carry off 5000 thermal units. The fans to be provided should therefore be proportioned to supply 300 cubic feet of air for each pound of steam condensed by the water to be cooled, so allowing some margin for non-saturation of the air; and there ought, in addition, to be a further margin of speed to be drawn upon when the atmosphere is considerably charged with moisture.

The weight of air to be provided is thus about twenty-five times the weight of boiler feed, or equal to the weight of circulating water, and the power absorbed in lifting this air to the top of the tower will be probably greater than that necessary to lift the water, owing to the smaller efficiency of fans as compared with water-pumps. It appears probable that the chimney type of cooler will ordinarily be better than the fan type for this reason; but it is to be remembered that the chimney type acts by temperature difference, and can never equal the fan type for efficiency, especially if very low temperatures are sought.

Cooling Reservoirs. — Where possible, a cooling reservoir should be placed so that the water will flow to it from the air-pump discharge, and where it will receive all roof and yard rainfall not required for feed purposes, otherwise an undesirable load may be thrown upon the air-pump. The jet condenser will raise its own water from the reservoir, a height which should not usually exceed 20 feet, and where a poor vacuum is expected this height must be reduced. It is well to provide a tank above the boilers, so that should all boilers be at rest on any occasion they can be filled by gravity.

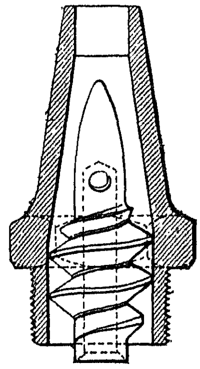


Fig. 1393

In the case of reservoirs used only by day, an average area has been found to be $33\frac{1}{3}$ square feet per I.H.P., the capacity being 198 cubic feet. For night and day working, an area of 32.4 square feet and a capacity of 255 cubic feet per I.H.P. have been given by Mr. Barker,[1] who found that with a surface of 18 square feet and a capacity of 99 cubic feet per I.H.P. with economical engines, fairly good results were obtained. He concluded that under the most unfavourable conditions in this country cooling may be as low as 190 B.T.U. per square foot of surface of the pond per hour, and it may rise to 290; while for really favourable conditions 600 units per hour may be dissipated. Hence about 1.8 square foot per 1000 units ought to be provided as a minimum, and a capacity of 10 cubic feet per 1000 B.T.U. to be dispersed; but great depth, if it can be got, is desirable. The paper named may be consulted in regard to cooling reservoir construction generally.

Spray Cooling.—The spraying nozzle, of which one form is shown in fig. 1393, as supplied by T. Ledward & Co., is fixed upon a system of pipes placed round the cooling-pond or tanks. They are often placed at a considerable elevation, the spray falling to the ground tank below inside a louvre wall, which is to prevent the blowing away of too much spray. With considerable pressure the jets break up more completely. Large numbers of nozzles are required to deal with the circulating water, as it is not desirable to work them at excessive pressures owing to the back pressures they produce on the pump. The annexed table gives their output for a head of 10 feet, the output for other pressures varying in proportion to the square root of the head approximately. Sprayers are unsuitable for use near a public road or house property, as it is difficult to confine all the spray within neighbourly bounds.

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[1] *Mins. of Proc. I. C. E.*, vol. cxxxii, part ii.

No.	Diameter of Orifice.	Diameter of Supply Pipe.	Approximate Discharge per Hour under 10 feet head. Gallons.
1.	$\frac{1}{4}$ inch	$\frac{1}{2}$ inch	90
2.	$\frac{1}{2}$ "	1 "	350
3.	$\frac{3}{4}$ "	$1\frac{1}{2}$ "	800
4.	1 "	2 inches	1400

Evaporative Cooling. — Where there is not room for sufficient pond area, and it is necessary to cool circulating water for re-use, the usual system is to employ large wetted surfaces exposed to the wind, or to a draught set up by a fan or by a chimney. The water is pumped to the top of the apparatus and descends by gravity, being distributed in thin films over the surface of thin boards, drain-tiles, or galvanized wire-mats. Some of the water is absorbed by the air passing by horizontally or vertically.

In the open-type evaporator a usual form consists of a number of thin boards, nailed at their top ends between two pieces of notched wood, which form at once a trough along which the water can run and be distributed to each vertically-suspended board by the notched edges of the trough. There are usually two lengths of board in the height of the tower, set at right angles to each other, so that whatever the direction of the wind one set or the other will be wholly edgeways to the wind, or both sets will be partially so, and allow the wind to blow through the narrow spaces between each line of boards. Water is distributed by notched troughs at the top of the apparatus, and is collected by other troughs and redistributed to the little troughs formed on the top of the lower tier of boards, finally descending to the pit or reserve below. Where sufficient area is given these open coolers will cool water below the atmospheric temperature. These open coolers are sometimes surrounded with a louvre frame or wall of wood. The louvres catch the water spray and prevent it being blown away except in small quantity. To cool 20,000 gallons per hour a ground space of 40 × 20 × 13 feet high should suffice. Mr. Arthur Koppel allows 208 square feet for 2000 gallons, 738 square feet for 20,000 gallons, and 2520 square feet for 100,000 gallons per hour. In his apparatus the drizzling boards are horizontal, and air passes both by virtue of atmospheric movement and partially by up-draught due to temperature.

Chimney Coolers.—The same apparatus as the foregoing, but provided with a chimney or closed shaft above, closely sheeted in with wood, is shown in fig. 1394. Here the up-draught is caused by actual chimney effect, and this is the more powerful as the water is supplied hotter, being so far self-regulating. As the chimney effect depends upon temperature difference, the cooling can never be so great as in an open-type apparatus, but it can be more certain, and, within its capacity, more efficient down to a certain moderate cooling effect.

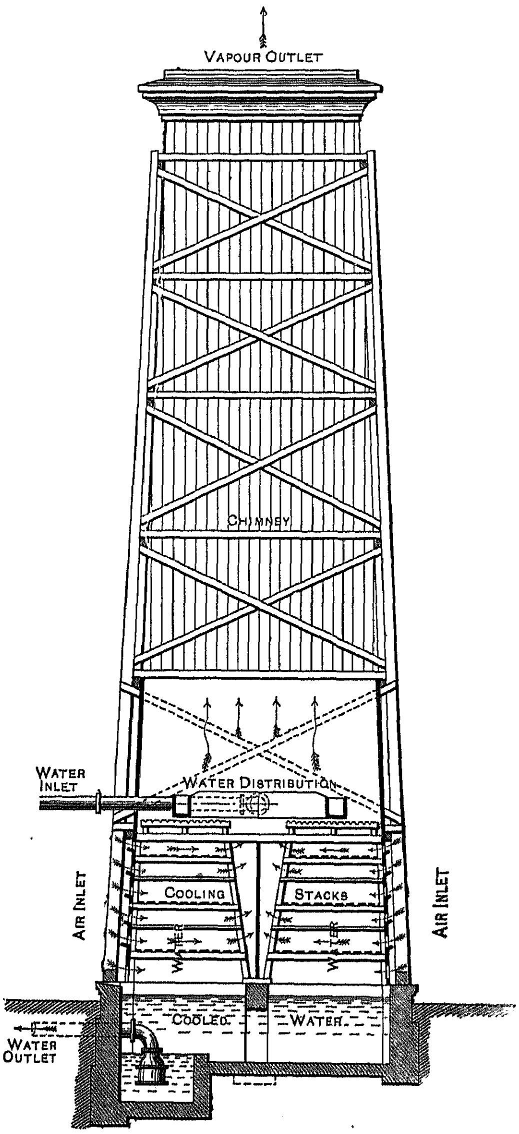

Fig. 1394.—Wooden Chimney Cooler—Balcke's System

Thus 2000 gallons per hour can be cooled on a floor space of 8 × 8 feet, the total height of chimney being 46 feet; 20,000 gallons require 19 × 19 × 57 feet; 100,000 gallons require 58 × 24 × 65 feet high; and 300,000 gallons can be dealt with in 164 × 24 × 80 feet high.

Such coolers require good foundations, and are to be calculated to withstand wind pressures. They should be kept away from buildings, as they throw off enormous volumes of steam and moist air, and damage brickwork if placed too close.

Fan Coolers.—Still more certain and effectual are the fan-cooled towers. These are similar to the foregoing, but have much less chimney, and air is forced through them by fans placed at the base of the tower. As they are not dependent on temperature for their effect they will cool water lower than the air temperature. Compared with the types already named, Mr. Koppel gives the following figures for fan coolers:—

20,000 gallons per hour, 11 feet × 13 feet floor space
100,000 " " 18 " × 24 " "

The brake horse-power of the fan motors is given as 2 for a water capacity of 7000 gallons, 5 brake horse-power for 20,000 gallons, and 25 brake horse-power for 100,000 gallons per hour.

One of these towers is shown in fig. 1395, constructed in this case of iron.

Chimney coolers may have chimneys of wood, iron or brick, and a concrete chimney is employed at Nuremberg to cool 77,000 gallons per hour, but timber is the usual construction.

As it requires power to drive the fans of a cooler, the fan cooler for condensing water will not pay to use where the space is sufficient for a chimney cooler.

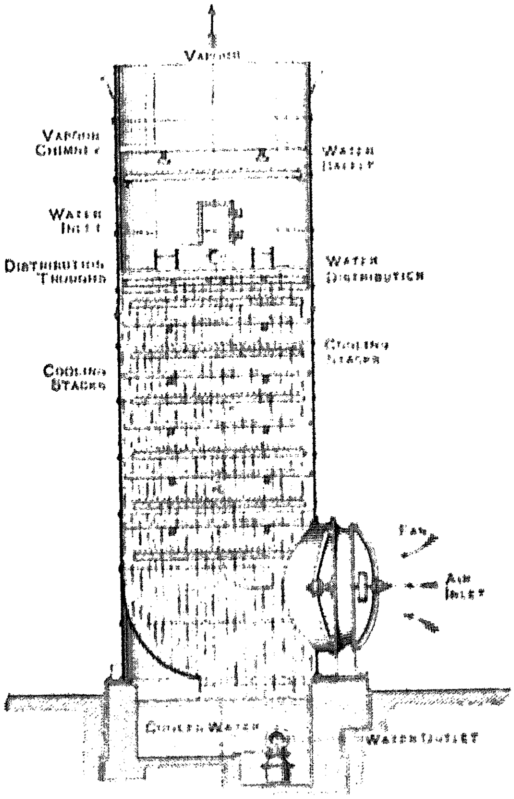

Fig. 1395.—Fan Cooler with Circular Iron Shell

The Worthington Tower.—This cooling-tower, fig. 1396, is filled with a succession of layers of drain-tiles stood vertically or with a similar succession of short metal tubes, the surface of which is made very great by splitting the tubes longitudinally and putting them together so that their circles interlace. As with mat towers, the water to be cooled is distributed over the ends of the top tubes by a revolving spreading-pipe or Barker's mill, and it is constantly changed and broken up as it reaches the next tier of tubes. Air ascends through the whole mass either by chimney action or by fan propulsion. The towers

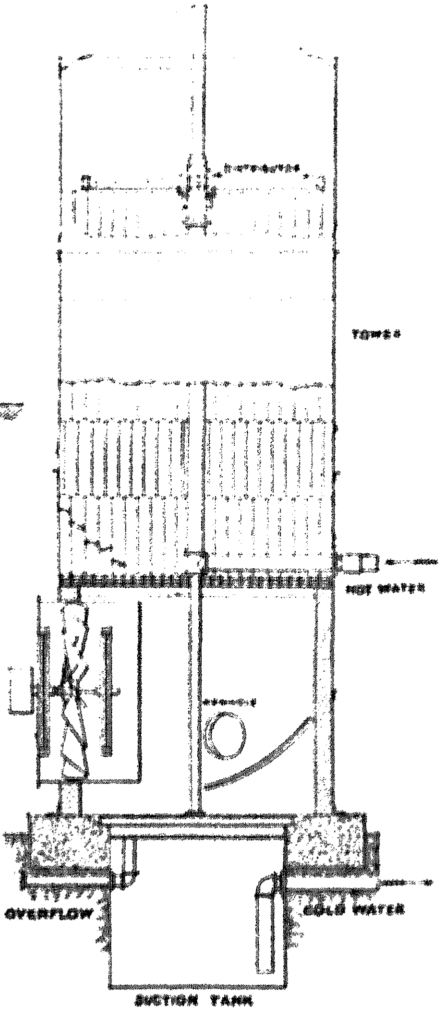

Fig. 1396.—Worthington Cooling Tower

are of steel plate, the glazed drain-tiles or galvanized split tubes are 2 feet long.

A natural-draught tower, occupying a total floor space of 30 feet × 30 feet, will cool the water to condense 30,000 lbs. of steam per hour, while a fan-cooled tower would stand on a space of $25\frac{1}{2}$ feet × 24 feet. These figures are for earthenware-pipe filling. For the metal-tube filling the floor spaces required in each case will be 26 feet × 26 feet, and 18 feet × $16\frac{1}{2}$ feet, for natural and fan draught respectively. Our illustration, fig. 1396, shows the fan tower, the natural-draught towers being similar, but higher.

Mr. J. H. Vail[1] has given figures of value relative to cooling-towers. The total amount of water evaporated was 115,587 lbs. per hour, and it was decided to add a condenser to the engine. A Barnard tower was selected of the twin type, each half 12 feet 3 inches × 18 feet × 29 feet 6 inches high, with two pairs of fans in all. The tower is of steel $\frac{3}{16}$ inch and $\frac{1}{4}$ inch thick, with suitable angle-bar stiffeners. Water is delivered by a 10-inch pipe extending the full length of the chamber. This pipe is slotted, and delivers to 96 distributing pipes also perforated. These deliver over 42 galvanized-wire mats made of No. 19 steel wire woven to a No. 5 mesh, each mat measuring 12 feet × 15 feet 6 inches, the total surface being thus 8064 square feet. The mats are hung vertically.

The four fans are each 8 feet diameter, and each equal to a discharge of 90,000 cubic feet of air per minute, at 150 revolutions per minute.

The rated capacity of each chamber is the cooling of water to condense 12,500 lbs. of steam per hour from 132° F. to 80° F., when the atmospheric temperature does not exceed 75° nor the humidity 85 per cent. The condenser discharge was 58 feet above the condenser, the water being circulated by a Blake vertical twin air-pump. The fans were direct-driven by small engines capable of varying speed to suit atmospheric conditions. The following figures were obtained under various conditions:—

	Jan. 31.	Feb.	June 20.	July.	Aug. 26.	Nov. 4.
Time	9 p.m.	8 p.m.	8 p.m.	8 p.m.	8 p.m.	5.35
Atmospheric temperature	30°	36°	78°	96°	85°	59°
Condenser discharge ...	110°	110°	120°	130°	118°	129°
Condenser suction ...	65°	84°	84°	93°	88°	92°
Temperature reduction	45°	26°	36°	37°	30°	37°
Fan speed, revs. per min.	36	0	145	162	150	148
Condenser vacuum ...	$25\frac{1}{2}$	26	25	$24\frac{1}{2}$	$25\frac{1}{2}$	25
Strokes of air-pump ...	30	30	37	44	43	28

On one day when the plant was run from 7 a.m. until midnight, with an atmospheric temperature varying from a minimum of 83° to a maximum of 103°, the condenser discharge varied from 106° to 128°, the suction from 91° to 98°, the fan speed from 140 to 160 revolutions per minute, with vacuum from 26 inches to 20 inches, and air-pump strokes from 50 to 38 per minute. The above figures are simply the extremes during the day, and not necessarily coincident. The power developed was 600 I.H.P. as a maximum, and 400 as a minimum.

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[1] *Trans. Am. Soc. M. E.*, vol. xx.

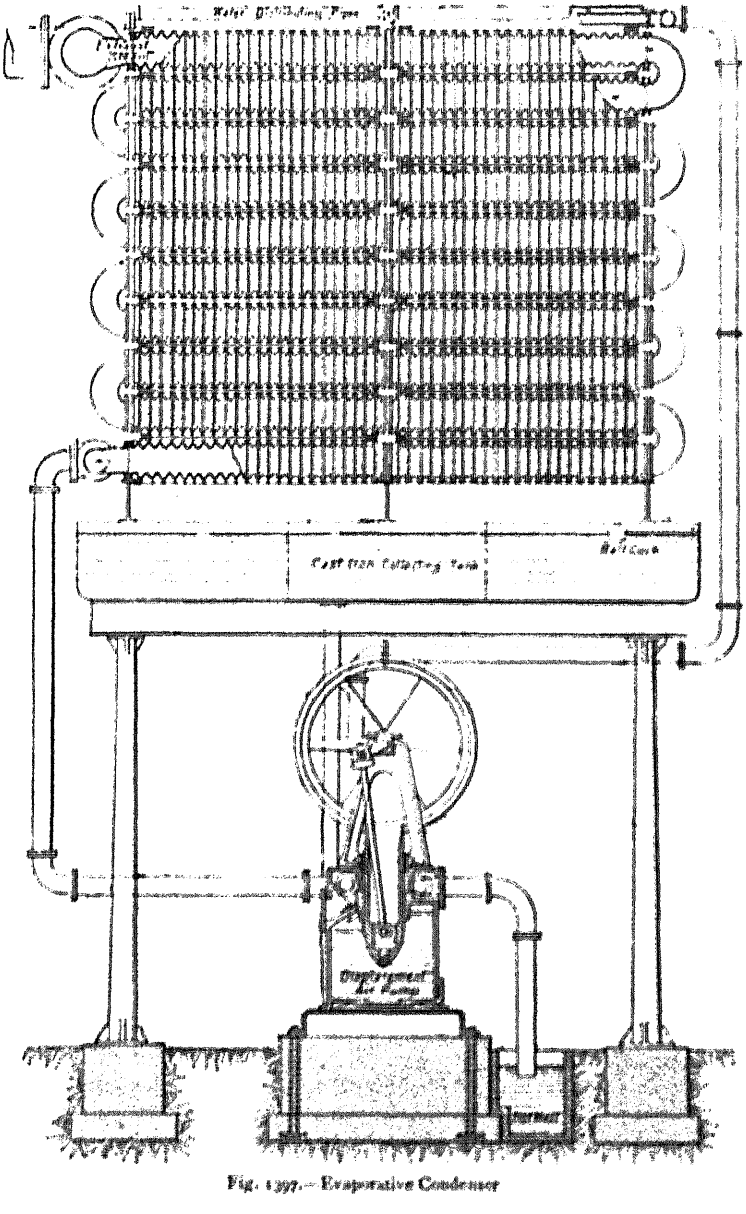


Fig. 1397.—Evaporative Condenser

On another occasion, in November, with a 25-inch vacuum and 643.3 H.P., there was 185.1 H.P. below the atmospheric line, so that there must have been at least 200 H.P. of work due to condensing. The work of driving the air-pump was 13.75 H.P., and the fan engines showed 13.5 H.P., leaving a balance of probably 173 H.P. from condensing plant, or about a fourth of the total power generated by the main engines.

The Evaporative Surface Condenser is a form of water-cooler inside which steam is condensed by the evaporative cooling effect of the atmosphere upon the thin films of water which are run over the gilled pipes of which the condenser is made. In this condenser the amount of water pumped over the pipes need not be more than ten times the weight of steam to be condensed. About two-thirds the weight of steam disappears from the cooling water, and the rest trickles off to the tank below and is re-pumped. These condensers, fig. 1397, are made by Ledward & Co. of London, and many are giving a vacuum of 26 inches. The small loss of water is, of course, partially due to the large radiating surface of the gilled pipes, which itself gets rid of a great proportion of heat apart from the evaporative effect.

The condenser serves the double purpose of a condenser and water-cooler.

From tests made by Mr. Longridge it would appear that an external surface of 30 to 40 square feet will condense 60 lbs. of steam per hour, with a vacuum of 24 inches.

CHAPTER V

WATER SOFTENING

The water ordinarily met with in steam engineering is very variable. Certain town supplies, such as Manchester, Leeds, Cardiff, and other towns, which obtain their supplies from the older rocks, including the millstone grits and the granite and Cambrian rocks, are practically pure, forming no scale in boilers and requiring no treatment except, perhaps, a slight amount of alkali to correct a slight acidity due to the presence of peaty acids. Water from the chalk, as Thames water, from the limestone rocks, from many gravels, on the other hand, is more or less charged with the carbonates of lime and magnesia; whilst water from the Keuper marls, such as occur at Burton-on-Trent, Newark, Leicester, and other places, is charged with lime sulphate, and produces a very hard and obstinate scale.

The carbonate salts owe their solubility in water to the presence of an additional molecule of carbon dioxide, which really points to these salts being bicarbonates, for monocarbonates are but very slightly soluble. This fact explains why it is that, when boiled, such waters are rendered soft, the extra molecule of gas being driven off and the simple carbonate salt rendered insoluble and deposited.

A simple carbonate may be represented by the formula for lime bicarbonate $CaCO_3 + CO_2$, or $CaO + 2\{CO_2\}$. This becomes $\{CaO + CO_2\} + CO_2$ when boiled.

If to the bicarbonate of lime there is added one molecule of caustic lime, or CaO, the following reaction occurs:—

$$\{CaO + 2[CO_2]\} + CaO = 2[CaCO_3] \quad \dots \quad \dots \quad (1)$$

The caustic lime has seized upon the loosely-attached molecule of carbonic gas and converted itself into carbonate of lime. Thus, by adding soluble lime to soluble bicarbonate of lime, there has been formed a double quantity of monocarbonate of lime, the whole of which deposits as sediment.

The atomic weight of the elements concerned are as follows:—

Calcium	= Ca	= 40	Magnesium	= Mg	= 24
Oxygen	= O	= 16	Magnesia	= MgO	= 40
Carbon	= C	= 12	Carbonate of magnesia	= $MgCO_3$	= 84
Caustic lime	= CaO	= 56	Sulphur	= S	= 32
Carbonic acid	= CO_2	= 44	Calcium sulphate	= $CaSO_4$	= 136
Carbonate of lime	= $CaCO_3$	= 100			

If on analysis a certain weight of carbonate of lime is found to be present in water, it can be removed by adding caustic lime in the proportion of 56 pounds of dry freshly-burned lime to each 100 pounds of the carbonate, and there will be formed 200 pounds of deposit, which may be dried and used as whiting. This is the Porter-Clark process, and can be carried out either by means of more or less elaborate continuous processes and appliances, or by means of two sufficiently large tanks to hold each a day's supply in which the process is alternately carried on.

The carbonate of magnesia = $MgCO_3$ is to be similarly treated, excepting that each 84 parts of magnesium carbonate demand 56 parts of lime to combine with it.

Lime sulphate, or $CaSO_4$, requires different treatment. This salt can be decomposed by sodium carbonate.

The following are the atomic and molecular weights of sodium and some of its salts:—

Sodium	= Na	= 23	Carbonate of soda	= Na_2CO_3	= 106
Hydrogen	= H	= 1	Sulphate of soda	= Na_2SO_4	= 142
Soda caustic	= $HNaO$	= 40	Water	= H_2O	= 18

If carbonate of soda be added to sulphate of lime in the proportion of 106 of soda to 136 of the lime salt, the result will be—

$$Na_2CO_3 + CaSO_4 = Na_2SO_4 + CaCO_3, \quad \dots \quad (2)$$

or 142 of sulphate of soda and 100 of carbonate of lime. In this reaction the two metals sodium and calcium have changed places, and the new salts are insoluble as regards the carbonate of lime, but exceedingly soluble as regards the sulphate of soda. In other words, in place of 136 parts of scale-forming sulphate of lime, there have been produced 142 parts of very soluble sulphate of soda, which will not deposit until the water in the boiler becomes very concentrated. Hence with sulphate waters care is needed not to work the boilers so long without blowing out as to cause the new soda salt to become concentrated to a dangerous extent.

Carbonate of lime may be removed by the use of caustic soda, $HNaO$, as well as by caustic lime. The reaction is as follows:—

$$CaO + 2\{CO_2\} + 2\{HNaO\} = CaCO_3 + Na_2CO_3 + H_2O.$$

Here each 100 of carbonate of lime = $CaCO_3$, requires 80 of caustic soda, and the result is that the extra molecule of carbonic acid gas, CO_2, which kept the lime carbonate in solution, is taken up by the caustic soda $HNaO$, and soluble carbonate of soda remains = Na_2CO_3. This indicates how a water may be treated which contains both carbonate and sulphate of lime. The addition of caustic soda will reduce the lime carbonate to the insoluble state, and the carbonate of soda thus formed will then attack the lime sulphate and reduce this to carbonate, which will deposit and leave sodium sulphate finally in solution. This double reaction may thus be made to economize soda and lime.

When the two scale-forming salts of lime are present in the ratio 100 of carbonate to 136 of sulphate, or, say, 6 : 8, then only caustic soda is required for complete treatment, for the caustic soda that deals with 6 parts of lime carbonate will be changed to exactly so much carbonate of soda as will deal with the 8 parts of sulphate of lime. If the ratio of sulphate of lime be greater than 8, then the additional parts must be dealt with by additional carbonate of soda, as in (2). If, on the other hand, the carbonate of lime exceeds the ratio 6, any excess may be best dealt with by adding caustic lime, as in (1).

Both caustic lime and soda must be kept in air-tight cases, or they will become carbonated by absorption of carbonic acid gas from the air. Where water to be treated contains organic impurity, it becomes difficult for sedimentation to take place rapidly. A little alum or iron alum, added with the other reagents, will coagulate the albuminoïd organic matters and promote the rapid deposit of all the lime salts formed.

The treatment process is also assisted by warmth, and deposit is accelerated by stirring up the old mud from the tank bottom. The old deposit forms a nucleus round which new deposit will crystallize in larger crystals which will deposit the more quickly.

The subject as above was first treated fully by Dr. Angus Smith, and the following brief rules were formulated by him:—

Water No. 1, containing carbonate of lime only.
  For every grain of carbonate of lime per gallon use 1060 grains of carbonate of lime per 1000 gallons, made caustic by means of 560 grains of caustic lime, or simply use 560 grains of caustic lime.
Water No. 2, containing sulphate of lime only.
  For every grain of sulphate of lime per gallon add 779.4 grains of carbonate of soda per 1000 gallons.
Water No. 3, containing both carbonate and sulphate of lime.
  (*a*) For each grain of carbonate of lime per gallon follow rule for No. 1.
  (*b*) Neglect the sulphate unless it exceeds 8 : 6 of carbonate.
  (*c*) For each grain of sulphate in excess of 8 to 6, act according to rule No. 2 for such excess.

Thus No. 1 treatment can be carried out wholly with lime, the use of carbonate of soda being supererogatory excepting under special circumstances. It was written as it is by Dr. Angus Smith probably because caustic soda was not then so easy to obtain as it is now, or perhaps because of the somewhat unsafe or disagreeable nature of that salt.

Where water is corrosive, it has usually no scale-forming salts in solution, or if it has, the corrosion of the boiler plates prevents scale from adhering. Acidity may be corrected by the use of carbonate of soda, or more cheaply by means of lime, which will often enable a thin protective coat of scale to be formed on the plates. Filtration through chalk or limestone chips will often serve to destroy any acid in the water.

As no water can be completely softened, the gradual evaporation of water in passing over cooling-towers concentrates the scale-forming salts, and such water must be run to waste from time to time or submitted to softening treatment, as it becomes hard to the extent of, say, 20 degrees, a degree of hardness being 1 grain per gallon of lime carbonate.

Various apparatus is made for the more or less continuous treatment of water. In one of these, the Archbutt-Deeley system, the softened water is impregnated with carbonic acid gas from a coke stove. The object of this is to recarbonate the unavoidable residuum of a peculiar form of lime —probably hydroxide—and prevent it from depositing in and choking pipes.

Solubility of Salts in Water.—100 parts of water will dissolve the following parts of various salts:—

	Cold.	Boiling.
Sulphate of lime	0.23	0.21
Carbonate of magnesia	0.02	—
Carbonate of soda	20.00	45
Caustic lime	.13	—
Carbonate of lime	.0036	—
Magnesium sulphate	31	130
Sodium sulphate	12	42

At still higher pressures many of the salts that are soluble in boiling water again lose their solubility, and modern high pressures have set up conditions which differ from those to which engineers were accustomed in days of lower pressures, notably the effect of small amounts of sea-water, the chlorides in which seem to decompose at high pressures and cause rapid corrosion. Every endeavour should therefore be made to provide pure water, or water that can be readily softened by simple chemical process, or heat, or a combination of these methods.

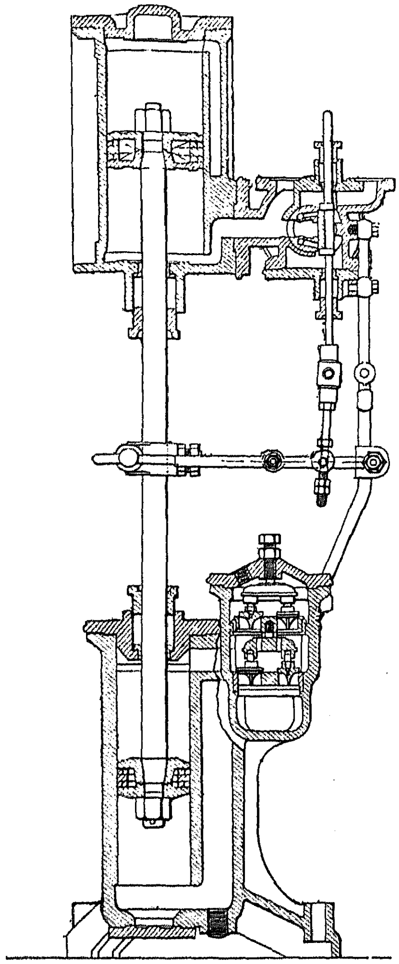


Fig. 1398.—Weir Feed-Pump

CHAPTER VI

FEED-PUMPS

The variety of feed-pumps is enormous. They exist as direct-acting steam-driven, with the steam and water ends tandem fashion; steam-driven, but with crank-shaft and fly-wheel, a class of pumps of which the Cameron is the type, and suitable for both boiler-feeding and fire purposes, and capable of working over a very wide range of speed and output; belt-driven pumps, in which the driven pulley is on a crank-shaft which drives one or more rams or buckets; electrically-driven pumps, which are of the crank type driven by belt or by gears, or by worm and wheel. Only a few can be illustrated.

The Weir Pump.—Fig. 1398 represents Weir's standard single pump, which is one of the solid slow-speed type of direct-acting pumps built for economy and reliability. Trials made with one of these pumps having 8-inch steam and 6-inch pump cylinder, with a stroke of 15 inches and a steam pressure of 107 and 110 lbs., gave the following results, and on twelve trials in no case did the pump efficiency fall below 95 per cent even at three double strokes per minute.

	Normal Speed.	Slow Speed.
Steam pressure at pump	110 lbs.	107 lbs.
Water ,, ,,	164 lbs.	164 lbs.
Revolutions per minute	15.9	6.0
Efficiency of pump	97.25%	96.6%
Pounds of water per pound of steam	84.6	55.3
Pounds of steam per net water H.P.	63.1	95.3
Foot-pounds work per B.T.U. to 212°	31.1	21.0

The exhaust from a steam-pump may be used either as a feed-heater, or it may be discharged into the intermediate receiver of a compound engine to do work in the low-pressure cylinder. The valve of the steam-cylinder is a D slide with a small auxiliary valve working on its back, and the arrangement is such that the pump will always work when steam is turned on. The barrel of the pump is of gun-metal, the rod of bronze, also the valves and their seats, the water pistons and the mountings, and these pumps will work up to 200 lbs. The annexed table will be useful as a guide to feed-pump provision, and the floor-space allowance for vertical pumps. The quantities in the table should be reduced by 5 per cent to allow for loss, which will be within that percentage.

STANDARD LIST OF SIZES AND DIMENSIONS OF THE WEIR DIRECT-ACTING PUMP FOR LAND INSTALLATIONS

Diameter of Pump.	Cylinder of Pump.	Length of Stroke.	Gallons Discharged per Double Stroke.	*Gallons Discharged per Hour at 12 Dble. Strokes per Min.	Floor Space.	Height.
ins.	ins.	ins.			ft. ins. ft. ins.	ft. ins.
5½	7½	15	2.49	1792.8	1 5½ × 1 7	6 4½
6	8	15	2.94	2206.8	1 9½ × 1 10	6 10
6	8	18	3.53	2541.6	1 9½ × 1 10	7 7
7	9½	18	4.86	3499.2	1 10 × 2 0	7 7
7	9½	21	5.67	4082.4	1 10 × 2 0	8 4
8	10½	18	6.352	4573.4	2 2 × 2 3	8 0
8	10½	21	7.43	5349.6	2 2 × 2 3	8 9
8	10½	24	8.49	6113.8	2 2 × 2 3	9 6
9	12	21	9.34	6724.8	2 4 × 2 6	8 9
9	12	24	10.67	7682.4	2 4 × 2 6	9 6
9½	12½	24	11.83	8521.2	2 4 × 2 6	9 6

* This is the best speed for boiler feeding.

In fig. 1399 is shown the Worthington Co.'s long-stroke vertical feed-pump, with 14-inch steam cylinders, 9½-inch pump plungers with 18-inch stroke.

The Injector.—The injector is a form of feed-pump which acts by the destruction of the volume of outflowing steam by the mixing of the steam with water, the combined jet then flowing with a velocity proportionate to the ratio of steam and water present. Thus a jet of steam may carry, say, five times its weight of water, the steam losing about 1000 heat units and the water gaining 200 units per pound. The combined jet would have a final velocity one-sixth that of steam. If the initial steam velocity was 1800 feet per second the final velocity would be 300, a velocity corresponding with a pressure of nearly 600 pounds.

When worked by exhaust steam the maximum ordinary pressure against which an injector will work is 70 to 95 lbs. for large sizes, and a supplementary jet of boiler steam is needed to help the jet against any greater pressure. The exhaust-driven jet has a temperature of about 190° F. as a maximum, but this is cold enough to condense the assisting jet of high-pressure steam. This is the whole secret of action divested of mathematics, the practical points in making an injector being the proportions to secure a firm, even, unbroken jet, the velocity of which, being greater than the velocity due to the boiler pressure, enables the jet to enter the boiler. The formula for velocity of flow of steam is $8\sqrt{\frac{n\mathrm{P}}{n + 1\mathrm{R}}}$, where V is the velocity in feet per second, *n* a

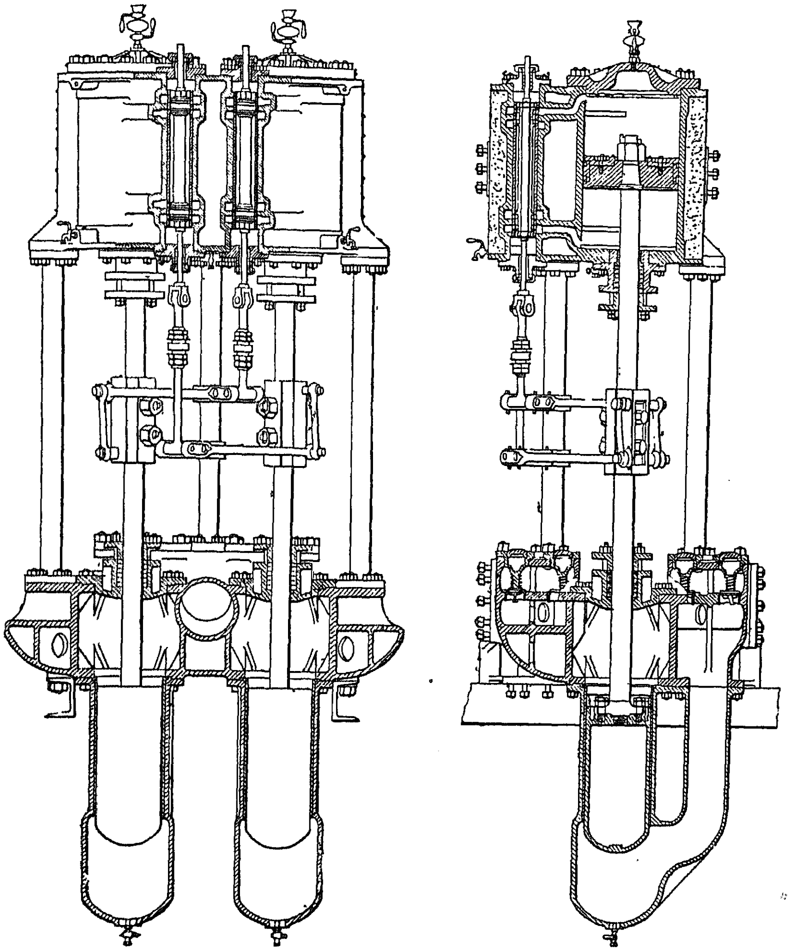


Fig. 1399.—Worthington Vertical Feed-Pump, 14″ × 9½″ × 18″

value varying from 1 to 1.4, P = the pressure per square foot, R = the weight of a cubic foot of the steam at pressure P, or

$$V = 8\sqrt{\frac{n}{n+1} P W},$$

where W is the volume in cubic feet per pound, n is the exponent in the term $P V^n = \text{constant}$. Approximately $V = 6\sqrt{\frac{P}{R}}$, the true figure being about 5 per cent less for wet steam and 2 per cent greater for dry steam or steam gas.

Injectors are made lifting, non-lifting, automatic restarting, and in very numerous shapes, but the principle of all is alike. All act on the principle of equal momentum. Consequently velocity of approach of water renders them more certain and effective. They form a useful stand-by feed for cold water, because they warm this, and they are also convenient and occupy small space. They should draw their water supply through a gauze sieve, as they are soon deranged by small obstructive particles. They are also apt to become scaled with incrustation, which can be dissolved off by weak hydrochloric acid.

Injectors are rated by the diameter in millimetres of the throat or jet, and will deliver about 1 gallon per minute for the 2-millimetre sizes, up to 36 gallons for the 12-millimetre size for steam pressure of 60 lbs., and about double these amounts for steam of 200 lbs. pressure, the delivery for different sizes varying as the square of the size in millimetres. Few injectors will lift over 18 feet, as a rule, with satisfaction. The water supply must also not exceed 75° for maximum capacity. An approximate rule to ascertain the output for any pressure is $(P + 85) D^2 = G$, where G is the pounds per hour, and P is the absolute pressure of steam, and D is the rated size in millimetres. For each foot of lift beyond 3 feet deduct 3 per cent from the delivery found by the above rule.

The discharge of exhaust-steam injectors appears to be $G = 170 D^2$.

Modified injectors can be used to warm and circulate water in feed-tanks either by use of live or exhaust steam. They are known as circulators, jet-pumps, ejectors, &c., according to form and use. One form is employed to create a vacuum to fill centrifugal pumps before these can start.

In fig. 1400 is shown a good example of a live-steam injector, the Sirius, by Holden & Brooke of Manchester. This injector is worked by one handle alone, which adjusts both the moving cones. If stopped by jerk or shock this injector is restarting automatically.

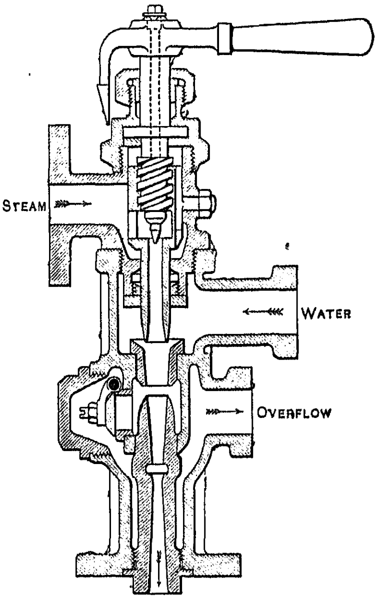


Fig. 1400.—Injector

Oil Separation.—One of the objections to surface condensation is the presence of oil from the cylinders. Oil is dangerous in boilers, especially when carbonate of magnesia is present, as it forms a light spongy mass that sticks to the furnace plates and causes them to become overheated.

It appears best to separate the oil from the exhaust steam on its way to the condenser. The De Rycke separator acts well. It consists of a series of spirally-placed blades in an enlarged part of the exhaust-pipe. These take out the water and oil, which drain into a receiver and may be discharged at intervals.

The De Laval Cream Separator has been tried with some benefit, but should be modified to suit the large flow of liquid and the small output of oil, which is the separated "cream". This machine acts by centrifugal action, which simply intensifies the separating effect due to difference of specific gravity of the oil and water. It is usually placed on the surface-condenser discharge. A long trough, with baffle or dip plates, would no doubt serve to separate oil from water. Probably a final saw-dust filter would improve the results of all separators, the difficulties of which are due to the extremely fine division of the oily particles and the consequent sluggish movement which they possess in water. Hence the benefit of the centrifugal separator, which, however, requires considerable power and care, as it is a high-speed machine and consumes steam or electric current to drive it.

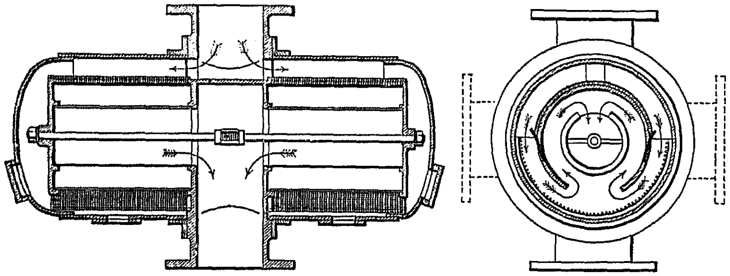


Fig. 1401.—Oil Separator

With a cooling pond or tower and jet condensers, the overflow of the water-tanks skims off the oil which rises gradually to the surface, and feed-water should be drawn from below.

Fig. 1401 shows the oil separator of Holden & Brooke, in which the exhaust steam is caused to reverse its direction of flow twice in flowing through the instrument, as well as being caused to move in a circle, whereby oil and water particles are caused to fly off into the wire-mesh lining, which holds the grease from being again picked up by the steam, and allows it to drain quietly away.

All separators act on the inertia principle, and embody the first law of motion.

The Davis-Perrett system of electrical oil separation from greasy water consists in passing the air-pump discharge through a tank filled with iron plates so connected that the potential across each element is 40 to 50 volts. A volume of 6000 gallons per hour is said to be dealt with at Leicester tramways station by means of ten series of plates, five in parallel across a 500-volt circuit. Six units of current are consumed per hour. The plates will last two to three years, and may be cleaned by reversing the current. The plates are so arranged that the water passes over their whole area. The effect of the operation is to cause the emulsified oil to coalesce so as to permit gravity effects to play. Sand and wood shavings are used as a filtering medium. The space occupied is about 100 square feet for 1000 gallons per hour to 290 square feet for 8000 gallons, or approximately $S = 100\sqrt{N}$, where S = square feet of area, and N = number of thousands of gallons per hour to be dealt with.

Section V

Miscellaneous Applications of Electricity

1. Electrochemistry and Electrometallurgy

CHAPTER I

ELECTROTHERMAL PROCESSES

Present Position and Scope of the Electrochemical Industries.—In the year 1800, shortly after the construction of Volta's pile was made known, Carlisle and Nicholson showed that the passage of the voltaic current through ordinary water effected its decomposition into hydrogen and oxygen. To give an historical account of the gradual development of modern electrochemistry would be out of place here. It is sufficient to say that the foundations of the subject were laid by Davy and by Faraday. Already, in 1807, Davy had succeeded in isolating the new alkali metals potassium and sodium, and the method then employed by Davy is substantially the same as that now utilized for the commercial manufacture of sodium. Faraday, besides discovering the principles which govern the construction of the magneto-electric machine, the prototype of the modern dynamo, was also the discoverer of the laws which bear his name.

Both Davy and Faraday seem to have recognized the importance of their investigations and the possibility of their industrial applications, for Davy (Bakerian Lecture of 1806) says: "It is not improbable that the electric decomposition of neutral salts, in different cases, may admit of economical uses"; while Faraday (*Experimental Researches*, 307) states that, "The capability of decomposing fused chlorides, iodides, and other compounds according to the law before established, and the opportunity of collecting certain of the products without any loss by the use of apparatus of the nature of those already described, render it probable that the voltaic battery may become a useful and even economical manufacturing instrument".

Before, however, the electric current came to be employed to any great extent in the electrochemical industries, a good deal of preliminary or pioneering work was necessary; and, since heavy electric currents are generally required, the galvanic batteries originally used had to be superseded by economical power plants and efficient dynamos.

Thus, if we neglect electroplating, and some other kindred industries such as copper-refining, in which a large amount of power is not required, it may be said that the electrochemical industries are not more than one, or at the most, two decades old.

The development of the electrochemical and electrometallurgical industries during this period, and their present position, may best be gauged by referring to the power now employed. In 1889, Borchers estimated that about one-quarter of a million horse-power was utilized or available, while according to Swan (*Journ. Soc. Chem. Ind.*, 1901, xx. 663–675), fifty works in Europe alone possessed 166,000 available horse-power. At the present time, therefore, it may be taken that throughout the world about one-quarter or one-third of a million horse-power is utilized or available in the electrochemical and electrometallurgical industries. Mr. Swan's estimate also shows that 89 per cent of the power in Europe is derived from water-power, 10 per cent from steam, and less than 1 per cent from gas-engines.

New electrochemical processes have, as a rule, to enter into competition with existing purely chemical processes, and in order to do so successfully they must possess certain advantages, the chief of these, other things remaining equal, being of course a lower cost of production. The following substances, for example, can be manufactured or treated both by purely chemical and by electrochemical means:—aluminium, sodium, copper, zinc, alkali and bleaching-powder, chlorates, phosphorus, the precipitation of gold from weak cyanide solutions, and the parting of gold and silver, &c. In some instances, notably in the case of aluminium and sodium, the electrochemical process has entirely superseded the purely chemical one. The same is almost true for phosphorus and chlorates. The electrolytic manufacture of alkali and bleach by the electrolysis of brine and fused salt has been so far perfected that it is now entering into competition with the Leblanc and ammonia-soda processes. It is, of course, not to be expected that electrical processes will replace all or even the majority of the older processes. All that can reasonably be hoped for in most cases is the application of electrochemical methods where the existing processes are imperfect, or where they fail entirely.

On the other hand, some electrochemical processes have the field entirely to themselves. This is the case when excessively high temperatures are required, or when the products are formed, and only formed, at temperatures higher than can be reached by the combustion of ordinary fuel. The electric furnace may then be employed, or recourse may be had to the high temperature produced by the combustion of aluminium ("thermite process"). Again, no competition is possible, even within the ordinary temperature range, when the products are obtained, and only obtained, by electrochemical means. Such is the case, so far as we know at present, with the alkali and ammonium persulphates, which are prepared by electrolysing solutions of the sulphates under the proper conditions.

Graphite, which is now produced in the electric furnace, forms an interesting example of a substance obtained by electrical means entering into competition with a naturally-occurring product. Peat coke or charcoal, manufactured in Norway in electrically-heated retorts, has to compete both with natural fuels and with wood charcoal made by the older processes.

Owing to limits of space, the foregoing must suffice to indicate generally the development, present position, and scope of the electrochemical and electrometallurgical industries. In what follows an attempt is made to compress into the space allowed an account of the general principles underlying modern electrochemical and electrometallurgical practice, together with descriptions, as far as these are available, of some of the more important industries and processes which are actually carried out in practice, and which illustrate the principles referred to.

Metallic and Electrolytic Conduction. — If we leave out of account the phenomena attending the so-called silent discharge of electricity through gases, about which we know very little at present, all electrochemical and electrometallurgical processes are based on the effects produced by a current of electricity in conductors. The transport of electricity through conductors is effected in two different ways, namely, by metallic conduction, and by electrolytic conduction. When a current is passed through a platinum wire, for example, the wire becomes more or less hot, but no material or chemical change can be observed in it. If the wire is now cut in two, and the ends introduced either into molten silver chloride or into acidulated water, the current of electricity will again flow through the system, but both the silver chloride and the water will be decomposed into their constituents, which separate out at the junctions between the wires and the electrolyte. Metallic conduction is unaccompanied by the motion of ponderable particles, while electrolytic conduction is essentially conditioned by the translation of matter, and is usually accompanied by chemical changes.

Metallic conductors, as the name implies, comprise all the metals, and certain other substances, such as graphite, peroxide of lead, &c., while electrolytic conductors or electrolytes include solutions of salts, acids, and bases, generally in water, and many chemical compounds usually in a molten condition.

The differentiation of conductors into metallic conductors and electrolytes is associated with the classification of electrochemical operations into electrothermal processes, or processes in which electrical energy is used solely for heating or smelting purposes, and into electrolytic processes, in which the primary effect is the chemical decomposition of the electrolyte. In the case of electrolytes, it is of course possible to combine electric heating with electrolytic decomposition.

General Principles of Electrothermal Processes. — In electrothermal processes the heat required for any chemical or metallurgical operation is generated by the passage of an electrical current of abnormal density through a suitable resistance. The resistance may belong to the class either of metallic or of electrolytic conductors, and since the object to be attained is merely the production of heat, without electrolysis, either a direct or an alternating current may be employed.

If heat only is generated, and no electrolysis takes place, it is easy to calculate the total heat developed by the expenditure of any given amount of electrical energy from the equation

$$H = 0.24\ C^2Rt \text{ grm.-calorie.}$$

As already mentioned, either direct or alternating current may be employed for heating. This equation refers to direct current, but is also applicable to alternating currents provided there is no difference in phase due to self-induction. In electric-furnace work there may be quite a considerable phase difference, which can, however, be obviated to a certain extent by installing alternators in which the frequency does not much exceed about 25 complete periods per second. But in any case the expression would become applicable by multiplying by the cosine of the angle of lag. The equation would then become

$$H = 0.24 \cos\phi\ C^2 R t \text{ grm.-cal.}$$

Having considered the relationship between electrical energy and heat, we may pass on to the principles involved in the practical application of electric heating.

In any conductor, or system of conductors in series, the current strength is always the same at every point quite independent of the material of which the conductor is made or of its cross-section. The potential along the conductor, on the other hand, falls from point to point. If the conductor is uniform there will be a regular fall of potential along its length. The electrical energy, *i.e.* the product of the quantity into the potential, similarly falls. In other words, electrical energy is consumed, and we get in its place an equivalent amount of heat. If the conductor is long the heat will be distributed over its entire length, and it may be that owing to the interchange of heat with the environs its temperature does not rise perceptibly.

When, however, we connect the ends of the cables from a generator through a short conductor, whose resistance is great compared with the resistance of the cables and of the generator, then practically the whole fall of potential occurs in the short conductor or resistance, and consequently the whole of the electrical energy is transformed into heat in it. This is the fundamental principle underlying the construction and use of electric furnaces. By applying it we can obviously obtain the whole of the heat corresponding to any quantity of electric energy whatsoever in a space which may be as small as we care to specify within, of course, the limits imposed by practical considerations. The temperature which is thereby obtained depends on a variety of circumstances, such as the mass of material immediately surrounding the heated conductor, its conductivity both for heat and incidentally for electricity, its specific heat, the time in which the heat is developed, &c. For example, a kilowatt-hour, when transformed into heat, yields 864,000 calories no matter whether the transformation occurs in a second or in a year. Practically, however, the effect produced would be quite different in the two cases. If the transformation took place in a second, the temperature of the resistance might rise to an enormous extent, while if it was spread over a year, there might be no appreciable rise in temperature at all owing to the radiation or conduction away of the heat as fast as it was produced.

By suitably altering the conditions, therefore, we can not only transform the whole output of a generator into heat in a small space, but we can also supply the heat at any temperature we please up to a certain maximum. We can also so dispose the resistance as to generate the heat exactly where it is required, which is often in a position otherwise difficult of access. When the heating of a mass of material is conducted from the centre we not only minimize the loss by radiation, but at the same time—and this is very important where high temperatures are concerned—we overcome most of the difficulties connected with the material, construction, and maintenance of furnace walls or retorts, and have not to contend with the great loss of heat which is inseparably connected with its transmission first through the walls of a retort and then to the interior of the material to be treated.

In spite of the fact that not more than about 10 per cent of the calorific value of fuel, when this is burned under steam-boilers, is again available for heating purposes in the electric furnace, the above advantages, among others, often make the costly electrical heating more economical in the long run than direct firing, even when the required temperature is within reach of the latter. When cheaper electric energy derived from water-power is to be had, there is, of course, still more to be said in favour of electrical heating. Now that large gas-engines can be constructed and power gas cheaply manufactured, it remains to be seen how far they will reduce the cost of electrical energy in large installations.

With regard to the electric furnace itself, this is merely a box or chamber containing a suitable resistance and the material to be treated. Provision has also to be made for charging in fresh materials, solid and gaseous usually, but sometimes liquid, and for removing material, solid, liquid, or gaseous, already acted on.

The suitable resistance, if only quite moderate temperatures are required, may be chosen from a considerable number of substances. Thus, in the electrical distillation of dried peat already referred to, a metallic wire spiral (probably iron), insulated with asbestos, is wound round the inside of the retort, and if required, also in the centre. For higher temperatures our choice of a resistance is strictly limited. We may either employ carbon, graphitized if desirable, the heated material itself, a gaseous resistance, or any combination of these. No matter which we employ the current, led up to the furnace by copper cables, is usually introduced into the furnace by carbon electrodes, poles, or terminals. In connecting these with the cables, as a rule by means of cast-iron or gun-metal clamps or holders, the surface of the holder bearing on the carbon should be at least equal to the cross-section of the carbon. A good contact is ensured by applying graphite paste.

Provided no chemical action occurs between the carbon resistance proper and the material to be heated, its temperature may be raised to about 3000° C. At higher temperatures it softens, and at about 3500° C., according to Violles' measurements, it evaporates or sublimes.

The material to be heated may also be used as the resistance. According to Nernst, the pure metallic oxides are practically non-conductors when cold. If heated, however, the conductivity increases, but only relatively slowly. On the other hand, the conductivity of a mixture of oxides increases rapidly when heated, and such a mixture of rare earths constitutes the "glower" of the Nernst lamp. This brings the behaviour of solid substances into line with liquids. Purified water conducts so badly that it is practically a non-conductor. Pure dry liquefied hydrochloric acid is also a non-conductor. When mixed together they form a good electrolyte. The fact that mixtures (or solid solutions) of solid substances, which are, if pure, themselves non-conductors, at any rate when cold, may become tolerably good conductors when sufficiently heated, should be carefully noted. In practice the raw materials used are seldom of such a high grade as to merit the term pure being applied to them, and the small percentage of foreign matter present may have a considerable influence on the electrical conductivity at a high temperature. Again, a furnace lining, when new, may be of such a character as to act as an insulator when heated as well as when cold, but after prolonged use it may absorb or dissolve foreign matter at a high temperature, and thus become a conductor and cause a leakage of current.

Lastly, the resistance employed may be a gas or vapour, as in all furnaces in which an arc is used.

The box or chamber containing the resistance and the charge may be constructed in a variety of ways, depending on the nature of the operation to be carried out. It may assume any shape. It may be constructed entirely of brickwork, or it may consist wholly or partially of a metal envelope lined with refractory material. It may be a permanent structure, made gas-tight, and if need be jacketed, or it may only be a temporary erection roughly put together, to be pulled down again in order to remove the contents.

The internal lining of the chamber is a matter of more importance. It should not be acted on by the heated charge. It should keep in the heat as much as possible, and act at the same time as a protection to the outer casing. In some cases it should also act as an electrical non-conductor. When this is not necessary, carbon is the material which is chiefly used. Sometimes fire-brick is all that is required. Magnesia bricks are also extensively used. The following general rule may be laid down:—If the material undergoing treatment is basic in character a basic lining, *e.g.* lime or magnesia, should be inserted, and fire-brick or silica in any form should be avoided. If, on the other hand, the material is acidic, some form of acidic lining, *e.g.* silica, should be used. The best lining of all, however, if this can be accomplished, is the material being treated. When this exists in the furnace in a molten condition a solidified crust acts as the best protection, and has the further advantage that by its use no impurities from the lining can get into the fused bath. On discharging the molten material the crust generally hangs together, and so prevents the outer walls from being exposed. If the heating is performed from the centre it is, of course, always possible to make the diameter of the furnace so large that the outer layers will remain comparatively cool, but in many cases, *e.g.* where a solid ingot is not produced, this would be found to be objectionable, for on discharging the furnace a mixture of material partly acted on and partly untouched would be obtained.

Types of Electric Furnace.—Although all electric furnaces are based on the conversion of electric energy into heat in a suitable resistance, it has been found convenient in practice to distinguish between resistance or incandescence heating, in which a solid or sometimes a liquid resistance is raised to incandescence by a current of abnormal density, and arc heating, in which the resistance is a gas or vapour. The distinction is somewhat arbitrary, and even in actual practice it would be difficult to prevent more or less arcing in what is ostensibly a resistance furnace, and conversely in some so-called arc furnaces, a portion of the heating, whether it is desired or not, is effected by the resistance method. If we look upon electric lamps as sources of heat instead of light—and this we are perfectly entitled to do, seeing that they only convert two or three per cent of the energy supplied to them into light while the remainder appears as heat—then the arc lamp would, of course, represent the type of arc furnace, while the ordinary incandescent or the Nernst lamp would illustrate the type of resistance furnace. Borchers (*Zeits. für Elektrochemie*, 1896, iii. 190) has characterized the difference between resistance and arc heating in the following way. Assuming the ends or poles of the main circuit to be equidistant, then in the case of resistance heating a low potential and high current strength would be required, while in the case of arc heating a high potential and low current intensity would be necessary.

Instead of classifying electric furnaces in this way it is open to anyone to adopt any other system of classification. For example, they might be divided into continuous working and discontinuous furnaces, or into direct and alternating, mono- or polyphase current furnaces, or into furnaces with one or more movable poles, &c. The system here adopted, however, is quite convenient, and taken in conjunction with the further subdivision proposed by Borchers, and about to be described, is believed to be of greater systematic value and to throw more light on the fundamental principles involved.

If space had permitted, it would have been interesting to trace the gradual development of the electric furnace. As this is impossible, we will have to be content to review the types of furnace from a point of view which is more or less arbitrary. The essential features having already been indicated, we have now to consider the mutual relations between the poles or terminals of the main circuit and the material to be treated, leaving further details to be described under the processes in actual operation.

We have already distinguished between resistance and arc heating, but each of these may be further subdivided into two distinct classes, so that we have altogether four different types of electric furnace, and it goes without saying, of course, that we may in actual practice have any combination of these.

In the first type of furnace the substance to be heated is itself placed in the circuit and made to act as the resistance, while in the second type, the substance to be heated is merely placed in contact with or adjacent to an electrically heated resistance.

In the third type of furnace the substance to be heated forms one or both poles of an electric arc, while in the fourth type it is placed in the neighbourhood of and heated by an arc.

The first three types are extensively used on the industrial scale, while the fourth has been equally extensively employed but chiefly for experimental purposes in the laboratory. Having chosen to adopt this point of view it would, of course, be easy to represent these types of furnace in a purely diagrammatic fashion. It is hoped, however, that the following illustrations will not only show the principles involved, but will also indicate to some extent how these are carried out on an industrial scale.

Fig. 1402 is a representation of one of the earlier forms of the Cowles furnace for smelting ores. It consists of a brick chamber A F, lined on

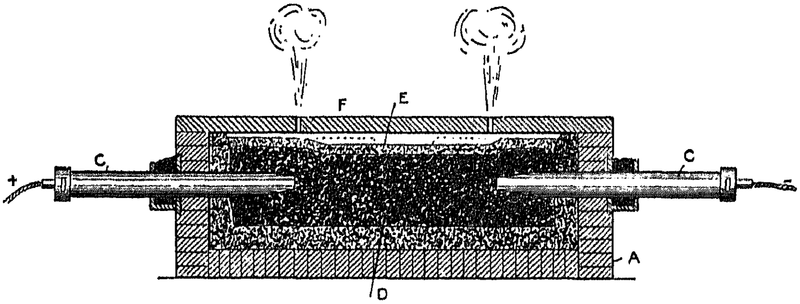
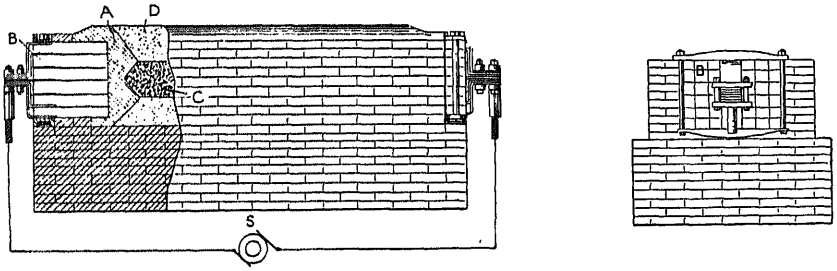


Fig. 1402.—Early Form of Cowles Furnace

the bottom, sides, and top with a protecting and non-conducting layer E, *e.g.* of charcoal, and enclosing the mixture D to be smelted, which in this case consisted of a pulverized intimate mixture of alumina and electric-light carbon in the proper proportions. C C are the main carbon poles or terminals short-circuited by the mixture only. The essential feature of the furnace is that the mixture, which conducts more or less when cold, conducts better when it becomes heated, but still remains a much greater resistance than the main carbons, cables, and dynamo, with the result that the Joule heat developed in it, according to the law already stated, is sufficient to effect the desired reaction. We are not here concerned with the possibility of the reduced metal acting further on the carbon. It is enough to know that the reduced metal does not form a regulus connecting the two poles, but filters downwards towards or through the lining, which is a good heat and electrical insulator. This furnace belongs therefore to the first type, in which the material itself acts as the required resistance. If in actual practice the conductivity of the cold material is so low that no current will pass when the main carbons are fully extended, they may at the start be brought closer together. Should this not be sufficient, they may be made to touch for an instant and then gradually drawn apart as the E.M.F. falls owing to the increased conductivity and cross-section of the heated mixture; or finally, they may be joined before the current is switched on, either by a thin metallic rod, which melts away as soon as sufficient heat has been developed to cause the material itself to conduct, or, if the metal is objectionable, by a very thin carbon rod, which may subsequently be dislodged or used up in the chemical reaction. Other alternative methods of starting up will, of course, suggest themselves, such as the initial application of a very much higher E.M.F. than is required in the actual operation.

The retorts used for distilling peat at Stangfiorden in Norway may be taken as an example of the second type of furnace. In this case, however, it is not so much a question of producing a high temperature as of generating the heat exactly where it is required. Similarly electric kettles belong to this type of furnace, the material to be heated, namely water, being placed in juxtaposition to an electrically-heated resistance. The best-known example, however, is the Acheson carborundum furnace.



Side Elevation

End View

Fig. 1403.—Acheson Carborundum Furnace

Fig. 1403 shows a side elevation, partly in section, and an end view of this furnace. The object to be here attained is the heating, to a sufficiently high temperature, of a mixture of sand and ground coke.

This is accomplished by placing the mixture D round a central core of coke C, which is raised to the necessary temperature by the passage of a heavy electric current. The resistance of the core can be adjusted by altering its length, cross-section, and the size of the grains of coke composing it. Good electrical connection between the core and the bundle of carbon blocks B, composing the main terminals, is effected by the mass A, of more finely pulverized carbon. A necessary condition for the successful working of the furnace is that the resistance of the core, whilst high enough to generate the required heat, is still less than the resistance of the surrounding material to be treated. In this way the path of the current is confined for the most part to the core, and does not spread out through the hot material being operated upon.

These two furnaces may be taken as representative of the two types of resistance, incandescence, or short-circuited furnaces. The Heroult furnace, to be described later under Aluminium, belongs to the first type. I add here another furnace, designed by King and Wyatt for the manufacture of calcium carbide. It is illustrated in fig. 1404, and according to the conditions it may be regarded as belonging either to the first or second type.

The main carbons E C are vertical, and are connected by a narrower carbon rod, whilst the material to be operated on is heaped up round it. If the carbon rod were chosen very thin, then on switching on the current it would be raised to an excessively high temperature, and either dissipated as vapour or dissolved up in the molten lime or carbide. After its disappearance the heated material would conduct the current, and it would act as a furnace of the first type. On the other hand, if the thickness of the connecting carbon were so chosen that it lasted throughout the run and conveyed the bulk of the current, the material being heated chiefly by radiant heat, then the furnace could be regarded as belonging to the second type; and obviously it might belong to both types at the same time.

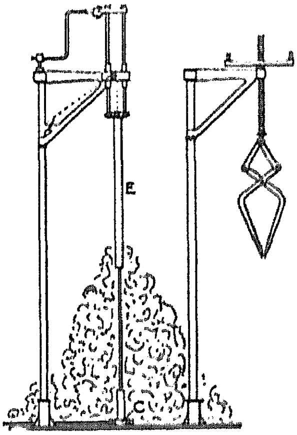


Fig. 1404.—Furnace for Manufacture of Calcium Carbide

Another reason equally important for noticing this particular type of furnace, which is not known to be in actual operation, is that it illustrates at the same time another of the essential features of the electric furnace, viz. the independence of the external shell of the furnace in relation to the operation effected. Siemens, to whom we owe so much as the pioneer of electric furnaces, stated in his paper read before the Society of Telegraph Engineers in 1880, "that the limit of heat practically attainable with the use of ordinary refractory materials is very high, because in the electric furnace the fusing material is at a higher temperature than the crucible, whereas in ordinary fusion the temperature of the crucible exceeds that of the material fused within it". In the case of this furnace proposed by King and Wyatt there is no furnace casing or crucible at all, and the inventors state that, in spite of the intense heat in the centre, the surface of the mound, except where the evolved gases are burning, is sufficiently cool to be handled readily. The suspended tongs shown in the figure are for removing the ingot of carbide prior to starting up again.

The two following types of furnace are distinguished by the use of an electric arc. Siemens, in the paper above referred to, described a furnace in which he melted steel, platinum, &c., and suggested that its advantages would make it a useful agent for carrying on chemical reactions of various kinds at temperatures and under conditions which it has hitherto been impossible to secure. The furnace employed by Willson (fig. 1405) for carrying out on the industrial scale the reduction of alumina and other refractory compounds is essentially a modified Siemens furnace.

The material to be treated lies in the crater or hearth of the crucible-shaped carbon terminal B, electrically connected through the iron sole-plate *b* with one pole of the dynamo, whilst the other is attached to the suspended carbon terminal C. The electric arc plays on and heats up the material, which thus practically forms one of the poles of the circuit. The crucible or lining B is contained within the masonry shell A, and is covered by the carbon or other plates E E, which are insulated from the crucible by the space *f* and the projecting masonry.

The fourth type of electric furnace is characterized by the presence of an electric arc in a closed space, while the material to be treated is not in contact with the carbon poles, but is heated by radiant heat only. Forms of furnace fulfilling these conditions are extremely useful for experimental work. Moissan, to whom we are especially indebted for his researches extending our knowledge of high-temperature chemistry, used for the most part a furnace of this description. His furnaces are modelled after the arc furnace described by Siemens in his patent 2110 of 1879. The chief advance made by Moissan in the constructional part of the furnace was the substitution of a lime or magnesia crucible or chamber for the carbon crucible employed by Siemens. In its simplest form Moissan's furnace consisted essentially of two blocks of lime, hollowed out and superimposed on each other in such a way as to leave an internal cavity into which projected horizontally the two carbon terminals. Between these the arc played, whilst the material to be treated was introduced into the lower part of the cavity. For details of Moissan's researches his book, *Le Four Electrique*, must be consulted. If the above definition were strictly adhered to it would be rather difficult to make an industrial furnace of any magnitude to comply with it. If, on the other hand, we merely understand by it a furnace in which a material, not necessarily itself a conductor, is in contact with or in close proximity to an arc, then some of the industrial furnaces could be comprised under this type. The possibility of using water- or air-cooled terminals, and of deflecting the arc on to the material to be heated, by means of a magnet, was also pointed out by Siemens.

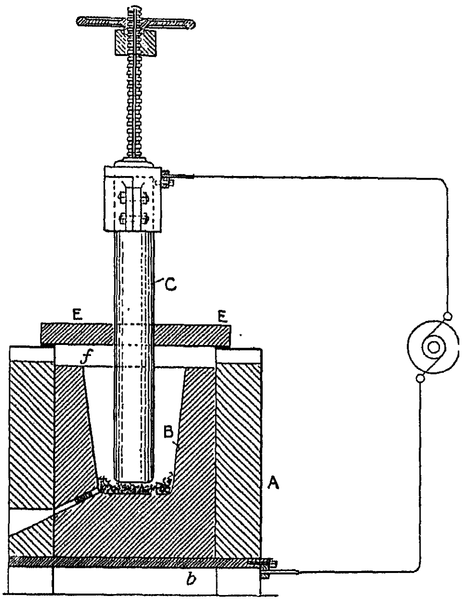


Fig. 1405.—Willson's Furnace

Direct and Alternating Currents.—Since all electrothermal processes depend solely on the conversion of electrical energy into heat, it would appear *prima facie* to be immaterial whether direct or alternating current is employed so long as a positive loss of heat energy does not ensue, owing to electrolysis taking place or otherwise. As a matter of fact, direct, alternating, mono- and polyphase currents are in practical use. In the case of three-phase currents, which are already used to some extent in calcium-carbide furnaces, the electrodes may either be coupled up according to the star or the mesh system, or each phase may be used in a separate furnace.

The question has sometimes been raised as to whether it is best to install direct or alternating currents. Each individual case would, of course, have to be separately considered, but in general it may be said that where there is a free choice in the matter alternating currents are employed. The question is capable of being looked at from two different points of view, viz. the electrical and the chemical. From the electrical point of view alternators would probably be chosen, as being simpler in construction than direct-current dynamos. They are, moreover, better able to withstand the sudden fluctuations of the current in electro-chemical work. From the chemical point of view the possible influence of electrolysis has to be considered, and whether it is likely to act favourably or unfavourably on the object to be attained.

It has been stated that when alternating currents are used the heat is more uniformly distributed in the furnace, and in particular cases higher outputs have been obtained by substituting alternating for direct current. In this connection it is well known that in an arc supplied with direct current the chief fall of potential occurs at the positive pole, and that consequently the bulk of the heat is developed there. Hence it follows that in a crucible furnace belonging to the third type the material to be treated, at any rate if not too deep, should form the positive pole. When alternating current is employed the terminals act alternately as positive and negative, so that we would expect a more even distribution of the heat between the terminals. This distribution of the heat might or might not be desirable, depending, for example, on whether the arc was maintained within a body of material to be heated, or only played on the surface.

The influence of self-induction in the case of the alternating current may also affect the distribution of the heat in the first and second types of resistance furnace. The tendency of alternating currents to travel by the skin of conductors is well known, and it seems not improbable that this may account for the better spreading out of the heat in furnaces operated by alternating currents, assuming, as has been stated by several authorities, that this actually takes place. The subject is one, however, which would repay further investigation.

Current and Power, or Watt Density and Temperature.—If no heat is to be wasted in the carbon terminals or electrodes, not more than a certain low-current density or drop of voltage must be employed. In the furnace itself, or rather in the part of it which is effectively heated, the current density is often an important practical guide to the temperature to be attained. Referring to the type of furnace shown in fig. 1404, we may suppose that two carbon terminals of large diameter are connected through a core consisting of a single thin carbon, as in Borchers' typical resistance furnace. If the cross-sections of these are 10,000 square millimetres and 67 square millimetres respectively, then on passing a current of 1000 amperes the current density in the thick carbons will only be 0.1 ampere per square millimetre, so that no serious heating ought to occur in them, whilst the current density in the thin core will be 15 amperes per square millimetre. Now, Borchers found that this is approximately the current density in the filament of an incandescent lamp, so that the thin core will be heated up to about the white heat of the filament, and the heat so produced will be communicated to the material which is packed round it. Borchers has made many experiments, usually of short duration, in such a furnace, and as the result of these he has published (*Zeits. für Elektrochem.*, iii. 219) the following figures, to give an idea of the chemical reactions which can be carried out with different current densities in the core:—

Current Density.	Remarks.
15 amperes per square millimetre	Temperature of filament of incandescent lamp.
10 „ „	All oxides are reduced.
4–5 „ „	Lime is reduced.
0.5 ampere „	Red heat scarcely attained.

These figures refer to a solid core of carbon, such as is used for arc lamps; but if the core consists of crushed carbon or coke, then they will no longer be directly applicable, since the weight of carbon in the unit of length will be less, depending on the size of the particles and the tightness with which they are rammed in. Borchers has therefore suggested that the current density should be referred, not to the cross-section of the resistance, but to unit weight (1 gram) of the conducting material in unit length (1 centimetre) of the resistance. As a rule, the conducting material is carbon in some form, and even when mixed with another substance to be treated, a knowledge of the current density required per gram of carbon present in each centimetre length to effect certain reactions would become a valuable guide in designing furnaces for other purposes and in estimating the power required. At present we have to rely almost solely on experiment, as sufficient data have not yet been accumulated for the purpose. The following provisional table by Borchers, and referring to experiments on a small scale, may be found useful:—

Diameter of Carbon.	Length of Carbon.	Weight of 1 cm. length.	P.D. per cm. of length.	Current Density per sq. mm. cross-section.	Watts Consumed per sq. mm. cross-section and 1 cm. of length.	For each Gram of technically pure Carbon in 1 cm. length.	
						Current Strength.	Watts Consumed.
mm.	mm.	grams.	volts.			amperes.	watts.
4	20	0.21	2.5	10	25.0	595	1487
4	30	0.21	2.3	8	18.4	476	1095
4	50	0.21	2.2	5	11.0	297	655
4	100	0.21	2.0	3	6.0	178	357
6	60	0.52	1.8	2.14	3.85	115	207
6	200	0.52	0.85	1.43	1.21	77	65
10	200	1.26	0.70	0.64	0.448	40	28
10	300	1.26	0.47	0.57	0.268	34	16

With a current density of more than 15 amperes per square millimetre (*i.e.* more than 900 amperes per gram of carbon in a length of 1 centimetre) very pure carbon rods became pliable, and the carbon began to volatilize, so that a temperature of about 3500° C. must have been reached.

Keller, in an address to the third acetylene congress in Paris, 1900, has also pointed out the insufficiency of a knowledge only of the current density in electrical smelting processes, and suggests the use of the term power or watt density, by which is to be understood the number of watts consumed per cubic centimetre of material traversed by the current.

As regards the carrying power of the main carbon terminals, Borchers so proportioned these that the current density should not exceed 0.1 ampere per square millimetre, or about 60 amperes per square inch. When, however, much heavier currents have to be employed, and especially when alternating currents of different frequencies are used, as is generally the case, a current density of 60 amperes per square inch would be certain to overheat the main carbons. Since alternating currents appear to be conveyed only in the skin or periphery of conductors, it is indeed questionable whether the carrying power of carbon terminals should be referred to the cross-section at all. A composite carbon terminal built up out of rectangular or cylindrical rods would have a very different carrying power from a solid carbon of the same cross-section, assuming that this could be made, depending on whether the separate rods were insulated from each other or not. These factors, and also differences in the carbons supplied by different makers, render it impossible at present to construct a table showing the maximum current that can be carried without overheating carbons of different cross-section or periphery. As a first approximation, however, for carbons intended to carry about 2000 amperes or more, it would be well to calculate on only about 20 amperes per square inch. Graphite terminals, having a conductivity three or four times as high as amorphous carbons, would, of course, carry a proportionally higher current.

There is a limit to the temperature that can be attained by electrical heating, just as there is a limit to the temperature that can be produced by the combustion of ordinary fuel in air. The latter does not greatly exceed 2000° C., owing to the dissociation of the products of combustion or the reversing of the reaction. In the former case the temperature is practically limited by the volatilization of carbon at about 3500° C. Although a temperature of over 2000° C. may be produced by direct firing, it cannot readily, without great loss of efficiency and severe destruction of building material, be utilized inside closed vessels. Even if 1500° C. or less has to be maintained within a retort, there is great difficulty in choosing refractory material of which to construct it.

The higher the temperature required, therefore, the greater will be the efficiency of electrical heating. Only in exceptional cases, however, is the temperature of the electric arc required. When possible, resistance furnaces are to be preferred, as they not only utilize the heat better, but the temperature can be more readily altered and controlled. Arc heating is essentially localized to the immediate neighbourhood of the arc, and whilst in resistance furnaces the heat may also be generated quite locally, there is ways the possibility of distributing it more or less uniformly over larger odies of material. Siemens as long ago as 1880 compared the efficiencies direct firing and electrical heating with current generated by the best eam-engines then known, and arrived at the conclusion that even for elting steel, which does not require an excessively high temperature, the ectric furnace may be considered more economical than the ordinary air rnace, and nearly equal to the regenerative gas furnace, as regards fuel onsumption.

Owing to the great differences in temperature at different parts of an lectric furnace, very little meaning would be conveyed by speaking enerally of the temperature of a furnace, and it is not possible in this rief account to discuss methods of estimating very high temperatures. nowing the amount of heat generated in a furnace from Joule's law, it ould be easy to calculate the temperature produced provided we knew ith any approach to accuracy the mass of material heated, the specific eats, the amount of heat absorbed or evolved during the chemical reactions, nd the losses of heat by radiation and otherwise. As we have to guess at ost of these the results are in no way satisfactory.

Continuous and Discontinuous Furnaces.—After the material has been ubjected to electric heating for the required time, it must be removed rom the zone of heat. This may be effected either in a continuous or n a discontinuous fashion. If the material fuses it may be tapped off hen necessary, and fresh materials fed into the furnace; or, whether it uses or not, it may, by suitable mechanical means, be passed continuously hrough the heated zone. When these methods are impracticable, the ontents of the furnace may be left to cool more or less before being discharged. Other things being equal, there can be no question that a furnace which works continuously, both as regards the treatment of material and the utilization of the electric energy, is to be preferred. Simple as the problem may appear at first sight, its solution presents considerable practical difficulties. Most of the furnaces at first used worked intermittently, as they were easier to construct, but now that the technical difficulties are being gradually overcome there is a greater tendency towards the use of continuously-working furnaces. A few typical examples of these will be described and illustrated under the manufacture of calcium carbide. At the same time it should be remarked that a furnace which is, strictly speaking, discontinuous, may form one of a set of furnaces or be part of a system which is to all intents and purposes continuous and as efficient as a single furnace working continuously.

Electric Smelting of Metals and the Production of Alloys.—Incidentally it has been mentioned that all the metallic oxides, when mixed with carbon, can be reduced when sufficiently heated. One of the first electric smelting operations to be conducted on the commercial scale was directed towards the manufacture of aluminium from its oxide alumina, Al_2O_3. When mixed with carbon and heated sufficiently alumina is reduced, according to the equation, $Al_2O_3 + 3C = 2Al + 3CO$. The metal so formed, however, is by no means pure. Even if produced from pure materials the metal at the high temperature required unites with carbon, forming a carbide, Al_4C_3, and in practice it was found advisable to add to the mixture another oxide or metal and thus produce an alloy which was free from carbon. When copper is employed for this purpose an aluminium-bronze, containing as much as 20 per cent of aluminium, can be obtained. Although it was originally intended to separate the aluminium from the alloying metal, the Cowles process resolved itself into one merely for the production of alloys. This became all the more necessary on the introduction of the electrolytic processes, which produce technically pure aluminium directly, and this can obviously either be used as such or melted down into alloys of the required composition.

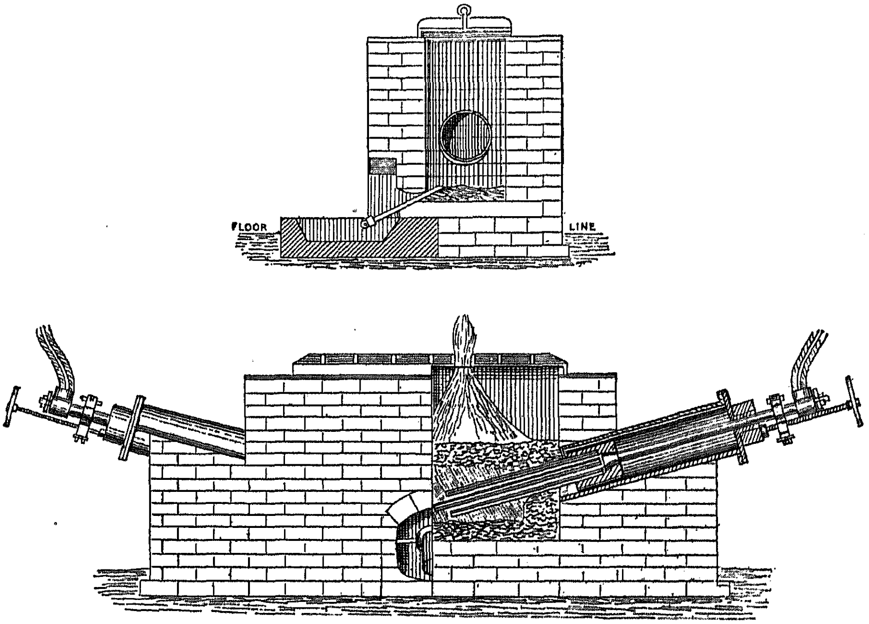


Fig. 1406.—Cowles Furnace

Nevertheless the introduction of the Cowles process marked the first step towards the introduction and use of the electric furnace for practical electrometallurgical purposes. In England the process was worked for a number of years at Milton, and although it is no longer in operation a brief description of the furnaces (*Journ. Soc. Chem. Ind.*, 1889, 678) may be of interest. As will be seen from figs. 1406 and 1407, they are an elaboration of the furnace shown in fig. 1402. They were built of fire-brick, the internal dimensions being 60 × 20 × 36 inches. A cast-iron tube built into each end admitted the terminals, which consisted of a bundle of nine 2¼-inch carbons or of five 3-inch carbons attached to a copper or iron holder. The position of the carbons could be adjusted by means of the screws shown in the figure. After covering the bottom of the chamber with ground charcoal, previously soaked in milk of lime and dried, the charge of alumina mixed with coarse charcoal and the metal to be alloyed was introduced, followed by a layer of coarse charcoal on the top. Some broken pieces of carbon arranged between the main carbons formed a bridge to start the current. By means of a copper rod and flexible cables the terminals were connected to the omnibus bars communicating with a dynamo capable of furnishing 5000–6000 amperes at 50–60 volts. After the furnace had been in operation for about an hour and a half the current was taken off and switched on to the next furnace. The melted alloy which collected at the bottom of the furnace was either tapped off or removed after cooling.

The amount of energy required per pound of aluminium contained in the alloy was high, being on the average 18 H.P. hour per pound. It may here be observed that an experimental run made with an alternating current instead of direct current gave the same results. Latterly the horizontal furnace was replaced by a furnace of the crucible type, having a bundle of vertical carbons as one of the poles.

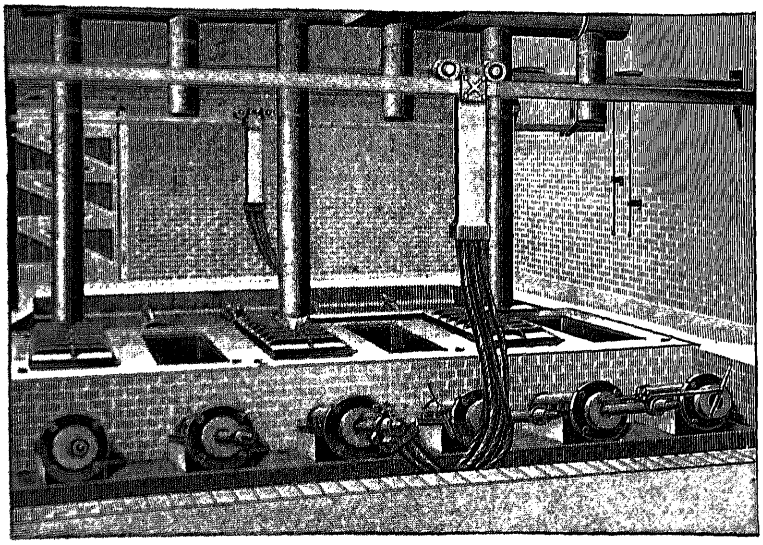


Fig. 1407.—Series of Cowles Furnaces

In contrast to the excessively high temperature which is necessary for the reduction of aluminium from its oxide, we have the comparatively easy reduction of iron from its oxides. Although this is conducted on an enormous scale in blast-furnaces by direct firing, experiments recently made near Rome showed that iron and steel, or alloys of iron with manganese, chromium, tungsten, nickel, &c., can be conveniently produced in the electric furnace. So successful were these experiments that the Stassano process is to be tested in Northern Italy with a larger plant, having 1500 H.P. available. The iron ore, mixed with the requisite amount of carbon and lime or silica, is made into briquettes and fed directly into the electric furnace. It is estimated that the electric energy required will be 3 H.P. hour per kilogram, or rather less than 1½ H.P. hour per pound of iron produced. Several other processes for reducing or treating iron in electric furnaces are now being tried on a commercial scale.

Cast chromium, molybdenum, and tungsten, &c., can also be produced in the electric furnace by smelting the oxides with carbon. When obtained in this way they almost invariably contain a certain amount of carbon existing as carbide. If special precautions are taken, and less than the quantity of carbon theoretically required for complete reduction of the oxide is employed, metals may in some cases be obtained practically free from carbon.

Since the principal application of these metals at present is in the manufacture of special steels, it is generally sufficient if they are obtained alloyed with iron. Ferro-chromium, ferro-silicon, ferro-titanium, &c., are already manufactured to a greater or less extent in furnaces similar to those described under calcium carbide, and can be produced in any quantity as soon as there is a sufficient demand.

When the pure metals are required, they may often with advantage be prepared by the Goldschmidt-Vautin or thermite process.

Calcium Carbide.—In attempting to prepare technically pure metals by the reduction of the oxides in presence of carbon at a high temperature, difficulties generally arise, owing to the fact that most metals combine with carbon to form carbides. Some of the carbides, notably calcium carbide, CaC_2, and silicon carbide, SiC, or carborundum, have, within the last ten years, become of great technical importance, and are at present exclusively manufactured in the electric furnace.

The development of the calcium-carbide industry has been responsible for the introduction of large numbers of special electric furnaces. The principles involved always remain the same, and the different modifications can all be reduced to one or other of the four types of furnace already considered. Many improvements have, however, been made, both in the system of working and in the actual construction of the furnaces, which permit the manufacture to be conducted more economically as regards consumption of electric energy, or reduction in the amount of labour, or more complete utilization of the raw materials.

The key to the manufacture lies in the fact that coke (carbon) dissolves in molten lime, with the evolution of carbon monoxide and the production of molten calcium carbide, according to the equation, $CaO + 3C = CaC_2 + CO$. Of course, a great many other circumstances have to be considered in the economical manufacture of carbide of good quality, but a complete discussion of these would occupy a small volume in itself. The equation gives the theoretical amounts of lime and carbon to be employed, viz. 56 parts of lime to 36 parts of carbon for the production of 64 parts of pure carbide. In practice allowance has to be made for the impurities in the lime and coke. The lime should be dead burnt, and should contain not less than 95 per cent CaO. It should also be as free as possible from magnesia, MgO (not more than 2 per cent), and from sulphur and phosphorus. Similarly, the coke ought to be of good quality, and should contain less than 7 per cent ash if possible. At first the lime and coke were employed in the finely-ground condition, and were sometimes briquetted. It is sufficient, however, and even preferable, according to Carlson, if they are simply crushed to pass a ½-inch or ¾-inch mesh.

This allows the hot carbon monoxide to diffuse more uniformly through the descending mixture, and thus give up its heat before escaping from the furnace. The use of the coarse mixture also diminishes the trouble arising from the finely-pulverized materials blowing out with the current of gas and from the formation of a bridge of sintered material between the terminals. The formation of such a bridge or shell imprisons gas, which ultimately causes a miniature explosion, whilst the colder material falling into the hot zone produces a sudden change in the resistance of the furnace.

Carbide furnaces are usually designed to act as arc furnaces of the third type, in which case an E.M.F. of 50–70 volts at the terminals is commonly employed; but they are also sometimes arranged to act as resistance furnaces, and this would appear to be the most reasonable type of furnace to employ for the purpose, as the temperature actually required is a long way short of the temperature of the electric arc. In practice, however, a compromise between the two methods of heating may be advantageous. In any case, overheating of the carbide made is nearly as bad as not heating the mixture sufficiently.

According to Rothmund (*Zeits. für Anorgan. Chem.* 1902, xxxi. 136), the temperature required for the formation of calcium carbide is about 1620° C. At a higher temperature the reaction will proceed more quickly and yield a more fluid carbide, but at a still higher temperature calcium carbide (and apparently all other carbides) dissociates into metallic calcium, and carbon in the form of graphite. The precise temperature at which the dissociation or destruction occurs is not yet known, except that it lies below the point at which carbon volatilizes, but the presence of graphite or of metallic calcium in the product indicate that too high a temperature has been employed.

One of the simplest carbide furnaces, as it was used by Willson, the pioneer of the carbide manufacture, is shown in fig. 1405. The provision of a tapping-hole indicates that it was originally intended to tap off the molten carbide. Owing to the high melting-point of carbide, however, this is an operation which presents some difficulties, and care must be taken not to overheat it, for the reason already mentioned. Whether or not a sufficiently molten pool of good quality and free from excess of lime is produced depends on the conditions of working. Anyhow, the furnaces first used for the commercial manufacture of carbide were arranged to work discontinuously. In this system of working, the vertical carbon, or group of carbons, was lowered on to the carbon hearth, and raised a little to strike an arc, and the materials were then fed in. As the molten carbide accumulates under the vertical carbon, and the resistance diminishes, the carbon is gradually raised, the arc being kept smothered. In this way an ingot or block of carbide not much greater than the cross-section of the carbon is gradually built up. When the latter has been raised as high as is practicable, the ingot is left to cool, and then removed. The first furnaces used at Spray were worked in this way, but in order to make the manufacture more or less continuous, the improved furnace shown in fig. 1408 was introduced. An iron car, the bottom of which was lined with carbon, and which served as one of the poles, was run into position in the brickwork chamber, and a block of carbide built up in it. It could then be removed and another substituted in its place. The unacted-on material surrounding the ingot acts as a protection to the iron truck.

The external appearance of another discontinuous carbide furnace, used to a considerable extent on the Continent, and designed and made by the Deutsche Gold und Silber Scheide-Anstalt, is shown in fig. 1409. The furnace chamber is lined with magnesia bricks, whilst the bottom of the chamber consists of a carbon block mounted on a trolley which is raised into position by gearing. The hearth is bevelled, and makes a gas-tight joint with the body of the furnace. The trolley and carbon hearth constitute one terminal, whilst the other is lowered in through the top of the chamber. Provision is also made for feeding in the material and for carrying off the gas evolved. This furnace is normally constructed to work with 2000–3000 amperes at 60–65 volts, or about 200 E.H.P.

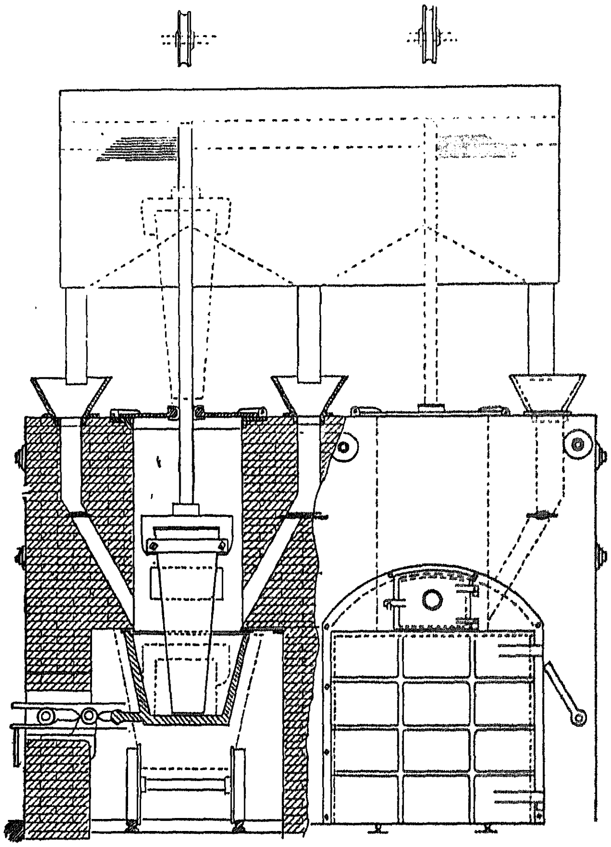


Fig. 1408.—Discontinuous Carbide Furnace

The crucible portion of furnaces, from which the carbide is tapped off in the molten condition, are usually thickly lined with carbon or magnesia, since the material itself can no longer act as a protection to the walls. Such furnaces are now also extensively employed. The furnaces used at Vernier, near Geneva, are stated to be cylindrical crucibles of compressed carbon 1500 millimetres in diameter and 800 millimetres high, provided with four tapping holes at different heights. The carbon is surrounded by a casing of boiler-plate, and the whole crucible is mounted on wheels. The vertical terminal consists of six carbon plates 1500 × 235 × 135 millimetres, weighing when new 390 kilograms. Each

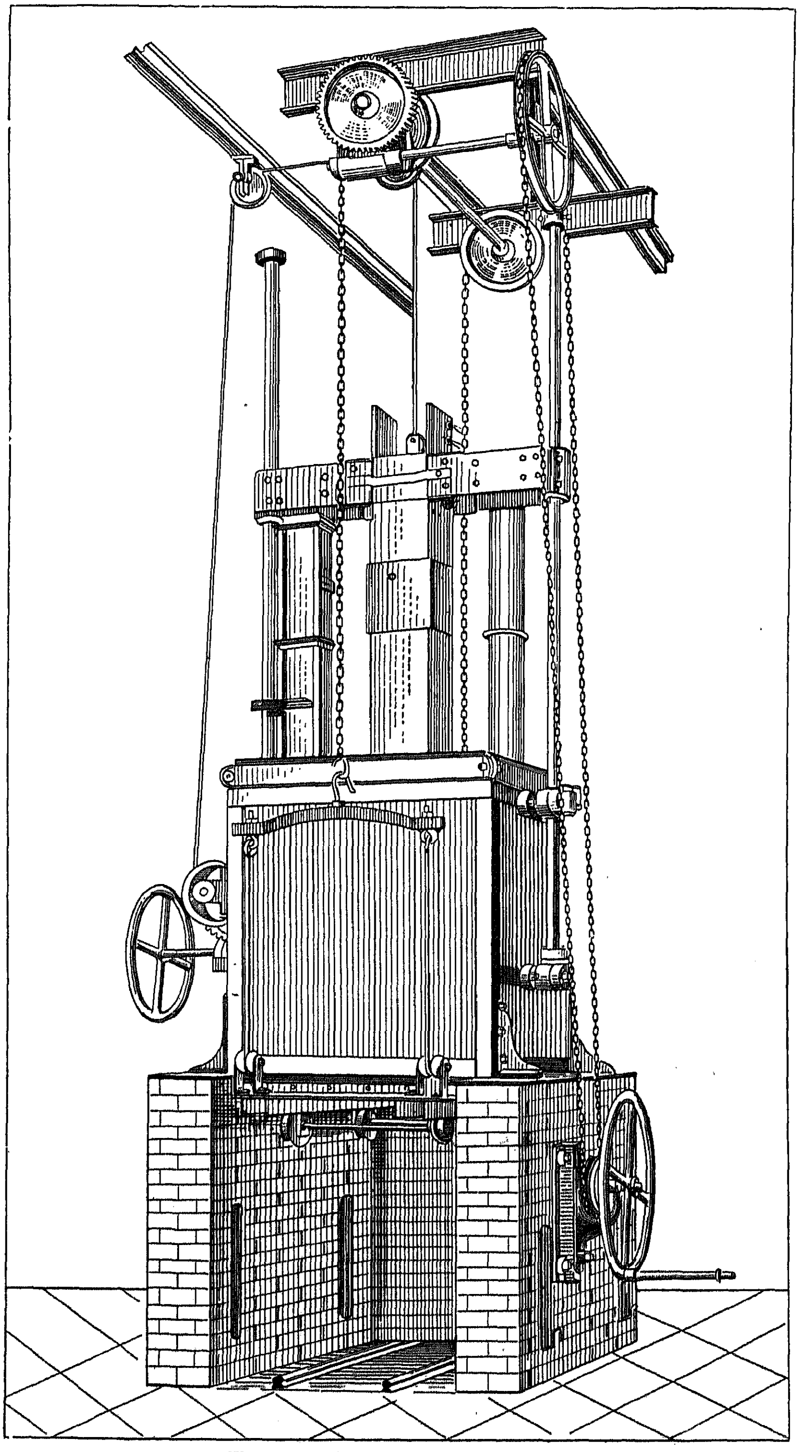


Fig. 1409.—Discontinuous Carbide Furnace

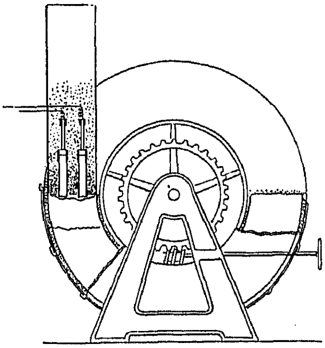


Fig. 141c

furnace requires 6000 amperes at 57 volts. The Rathenau furnace, or a modification, is also extensively used on the Continent.

Besides the type of continuous furnace in which the carbide is removed in the molten state, there are other types in which it is continuously withdrawn in the solid condition. In one modification of the Horry furnace, which is said to be in use at Niagara Falls, and Sault St. Marie, Michigan, this is accomplished as shown in fig. 1410. The pool of molten carbide formed between and below the terminals solidifies as the receptacle is slowly rotated, and as it descends plates are bolted on to the periphery to retain the carbide. When it arrives at the opposite side, those plates are again removed and the carbide broken out so that the formation can go on continuously.

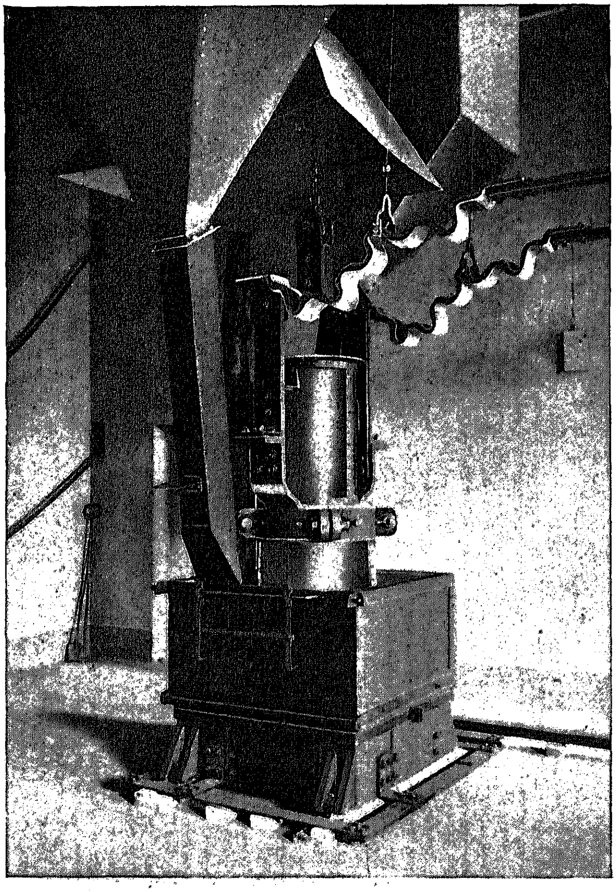


Fig. 1411.—Siemens and Halske Furnace for making Block Carbide

Siemens and Halske have constructed various types of furnace both for continuous and discontinuous working. A special feature of their furnaces is that the main vertical carbon is hollow, to allow of the escape of gas, the mixture being fed in round it. The hollow carbon should not affect the carrying power for alternating currents. Fig. 1411 shows the external appearance of one of their furnaces for making block carbide.

A great deal has been said both for and against continuous and discontinuous systems, but a discussion of the question would lead us beyond the scope of this article. Both, or rather all three, of the above systems are in actual operation, and it must be remembered that any given system, even although it is theoretically inferior to another system, may in capable hands give better results both as regards quality and output of the product than is obtained with a theoretically better system in inefficient hands. The quality of the product can readily be inferred by the amount of acetylene gas it yields on treatment with water. A good carbide should give at least 300 litres per kilogram, or about 5 cubic feet per pound. The energy consumed in its production, on the other hand, is not so easily estimated. It is true that the readings of an ammeter and a voltmeter, a wattmeter and a phasemeter, present no difficulties neither does the weighing up of the packed product; but the exact calibration of most of the meters for heavy electrical currents is not altogether an easy problem. Again, when locality affects the discussion, it is almost hopeless to lay down any hard-and-fast rule. Among other items, electric energy and manual labour are required, and the cost of both of these varies in different districts. If, in addition, the relative amounts expended for energy and labour also varies in the different systems, in order to produce a given quantity of packed carbide of the same or of a different quality, the difficulty of the problem may be estimated.

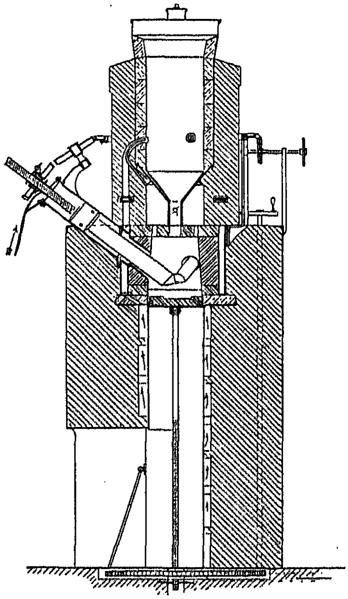


Fig. 1412.—Memmo Semi-Continuous Furnace

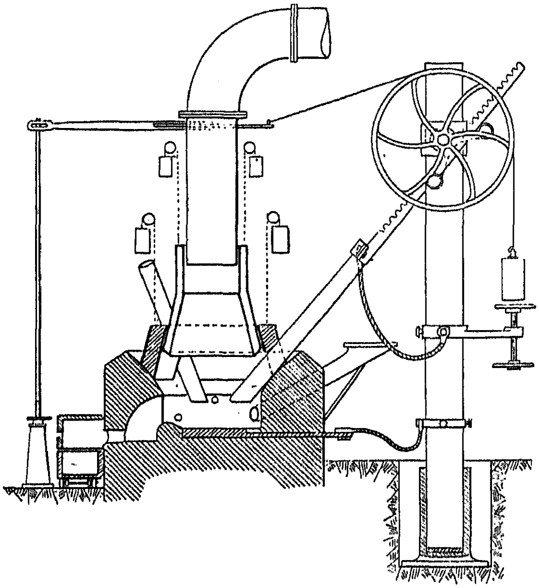


Fig. 1413.—Bertolus Furnace

The Memmo semi-continuous furnace, arranged for three-phase currents, is represented in fig. 1412. When the three carbon poles are connected up in star system, any reduction in the E.M.F. can be compensated by lowering the position of the carbon hearth by means of the gearing shown. If, however, they are arranged according to the mesh system, the position of the hearth or of the pool of fused carbide will have no influence

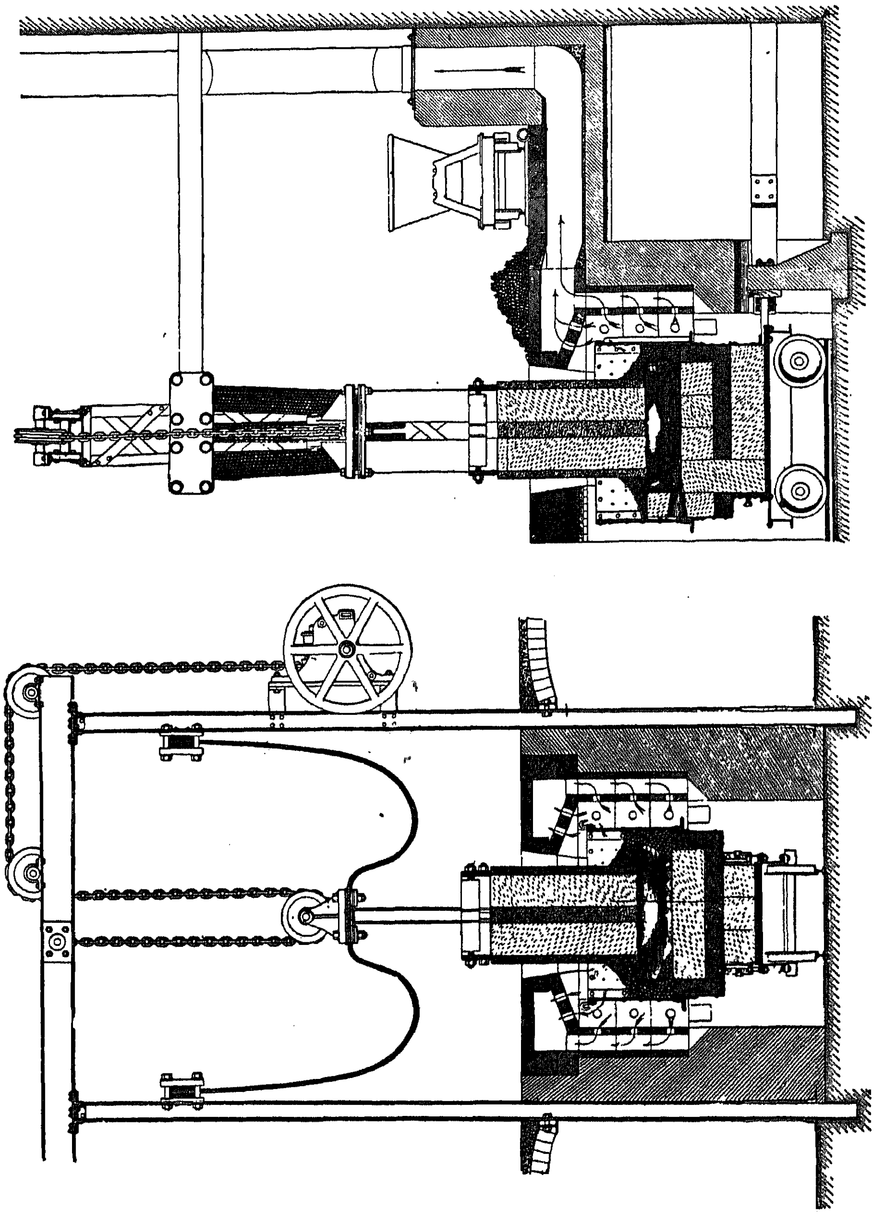


Fig. 1414.—Gin and Leleux Furnace

on the E.M.F. A more even distribution of the heat is to be expected with polyphase currents, and in the latter system no loss of voltage ensues owing to the resistance of the block of carbide formed. When the hearth has been lowered to the bottom, the carbide is taken out, the remainder raised, and the operation restarted. The Bertolus furnace is also operated with three-phase currents, but in this case, as is shown in fig. 1413, molten carbide is drawn off continuously.

The Gin and Leleux furnaces, also extensively employed, are arranged to work not only continuously, but also as resistance furnaces. The general construction of these furnaces is obvious from fig. 1414, whilst fig. 1415 shows the external appearance of a bench of these furnaces, of 300 kilowatts capacity, installed by Keller, Leleux, & Company, of Paris.

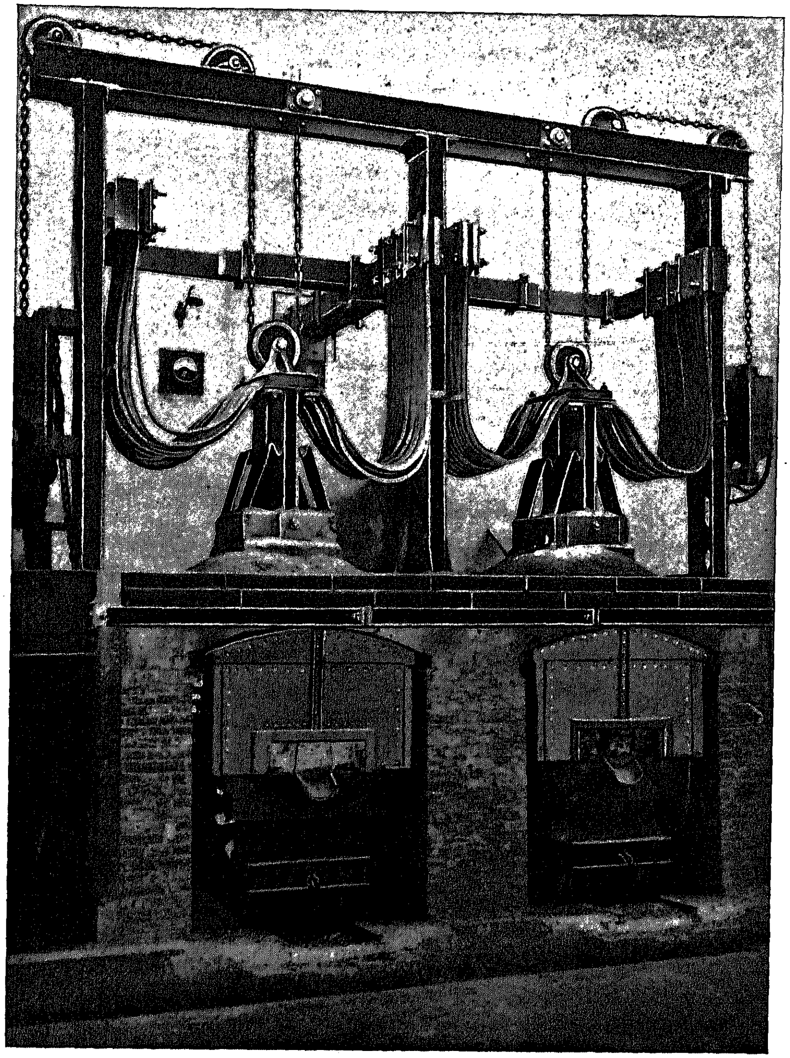


Fig. 1415.—Gin and Leleux Furnaces, 300 kilowatts capacity, by Keller, Leleux, & Co., Paris

The main vertical carbons of high conductivity are protected from oxidation by a skin of less dense carbon of inferior conductivity, while the carbon bed carried on the truck is of peculiar construction, and is so arranged that the upper part of the bed is separated from the lower by concrete or other non-conducting material. Communication between the two, however, is established by a carbon of smaller cross-section, which is placed adjacent to the tapping hole. The heat developed in it thus prevents the carbide from solidifying. Should it do so by any chance, provision is made for building up an ingot of carbide to the full extent, and for removing it before restarting.

It is hoped that these examples will be sufficient to indicate some of the more important modifications which the carbide electric furnaces have undergone.

The amount of electrical energy required in the production of calcium carbide has, of course, been variously estimated by different authorities, depending no doubt on the system of furnace employed, the calibration of the measuring instruments, and on whether the carbide was weighed up in the untrimmed or in the packed condition. The figures usually lie between 4 and 5 kilowatt-hours per kilogram, or, say, 4000–5000 kilowatt-hours per ton. Since an electrical horse-power year of, say, 300 days, is equivalent to $300 \times 24 \times .746 = 5361$ kilowatt-hours, it may be taken that at least a ton, or perhaps a ton and a quarter of commercial carbide can be produced per electrical horse-power per annum. As the cost of the latter may be anything between, say, £1 and £10, and as the selling price of carbide is about £12, it will be seen that the cost of electrical energy is an important item in the cost of manufacturing carbide.

At various times figures have been published which are supposed to represent, approximately at any rate, the theoretical amount of energy required to produce a given weight of carbide. For the reason given on page 247, such figures are of no value whatever except in so far as they show that the energy actually consumed is of the same order of magnitude. It is thus impossible to state with any approach to accuracy the energy efficiency of the electric furnace in the manufacture of calcium carbide.

Pure calcium carbide is a colourless crystalline substance, but the ordinary commercial article, containing about 80 per cent CaC_2, is usually a reddish or black crystalline body, depending on the conditions of manufacture. The European production in 1900 was stated to be 60,000 tons, with facilities for producing three times this amount if required. It is chiefly used for the generation of acetylene, but attempts are being made to introduce it as a reducing agent in metallurgical operations.

Carborundum.—Silicon carbide, SiC, or carborundum, is a rather brittle crystalline compound only exceeded by the diamond in hardness. It is manufactured by heating a mixture of silica and carbon (sand and coke) to a high temperature in the Acheson electric furnace, according to the equation $SiO_2 + 3C = SiC + 2CO$. To the mixture is added a small proportion of common salt and some saw-dust, to render it more porous and thus facilitate the escape of the gas. The resistance type of furnace used is shown in fig. 1403. The end walls, 2 feet thick, are the only permanent part of the furnace. In the centre of each of these 60 carbon rods, 30 inches long by 3 inches diameter, are attached to a cast-iron frame connected by cables with the main copper conductor, having a sectional area of 8 square inches. The internal dimensions of the furnace are about

16 × 5 × 5 cubic feet, the side walls being loosely built up every time the furnace is charged. The central conducting core consists of 1100 lbs. of new carbon, or of 850 lbs. of old more or less graphitized core, about 21 inches in diameter and 14 feet long. Round this is packed the mixture to be heated. A transformer of 830 kilowatts capacity transforms a 2200-volt current into one of 185 volts, and by means of a regulator the voltage can be raised to 250 or lowered to 100.

About an hour after switching on the current 1000 horse-power is being expended in heating up the furnace, and as the resistance diminishes the E.M.F. is regulated accordingly, until finally the resistance becomes constant. A run occupies about 36 hours. Fig. 1416 is a reproduction of a photograph showing one of the furnaces in operation. After cooling for a few hours the side walls are taken down and the unaltered mixture is removed. Radiating from the core for a distance of 10 or 12 inches are the beautifully coloured crystals of carborundum, which in some places attain a considerable size, and beyond this are layers of amorphous carborundum. The yield of crystalline carborundum per furnace per run is about 7000 lbs., and the energy expended somewhere about 4 kilowatt-hours per pound. As in the case of calcium carbide we cannot at present with any degree of accuracy state the energy efficiency of the carborundum furnace.

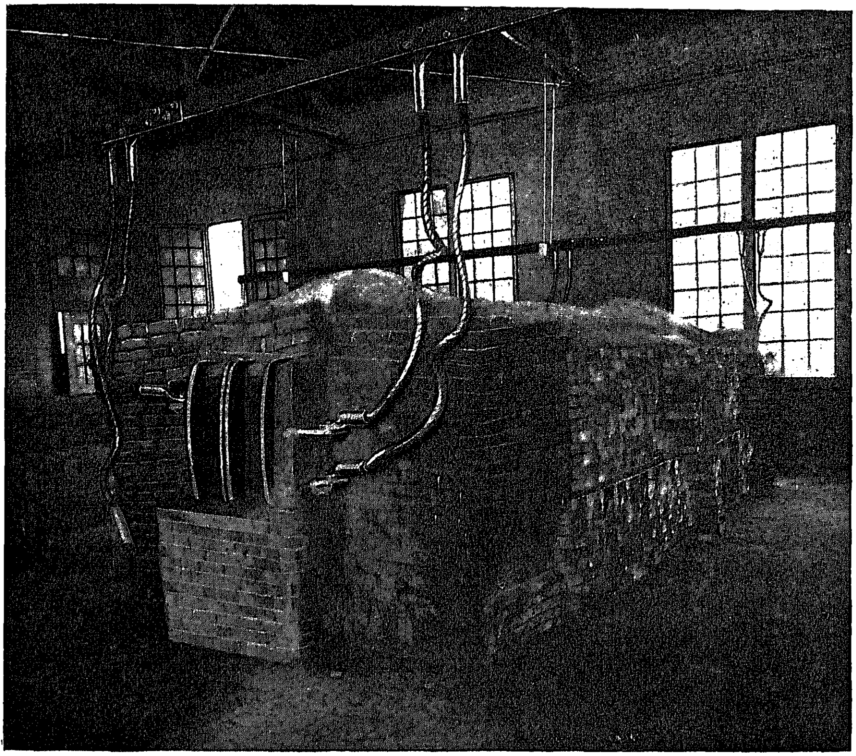


Fig. 1416.—Carborundum Furnace in operation

The present output of carborundum from the Niagara works is at the rate of 3000 tons per annum, and of this about one-half is used for abrasive purposes and one-half in metallurgical operations. In 1897 the output of carborundum was only about 600 tons, so that the increase in the rate of production speaks for itself.

Graphite.—Along with carborundum, graphite is another product of the electric furnace the manufacture and introduction of which is chiefly due to Acheson. Naturally-occurring graphite is nearly always contaminated by the presence of foreign and undesirable matter, whilst the graphite manufactured by electrical means is practically pure. It was stated that all carbides when overheated yield up their carbon in the form of graphite. At the temperature necessary for this destruction of the carbide, the other element associated with the carbon to form carbide is dissipated in the state of vapour. This is the principle underlying the manufacture of graphite in the electric furnace. It is, of course, impossible to enter into all the details here. Acheson observed, however, that it was unnecessary to first combine the substances in the proportions required by the chemical formula of the carbide, and then to effect the dissociation. A relatively small quantity of the carbide-forming constituent, properly distributed, is all that is required, owing to the catalytic or contact action of this constituent. Anthracite coal containing a suitable proportion of ash can be completely converted into graphite on heating in the electric furnace, whilst other varieties of purer amorphous carbon, such as oil coke, cannot. This can be explained by the progressive formation and decomposition of carbide.

Carbons similar to those employed in electric arc lighting, but containing a suitable proportion of impurity, can also be converted, on strongly heating by a current of abnormal current density, into almost pure carbon in the graphitic form.

Graphite produced in the electric furnace can, of course, be used for all purposes for which the naturally-occurring substance is now employed, and is in fact to be preferred on account of its greater purity and uniformity.

The graphitized carbon terminals and anodes, which are already extensively employed for electrochemical and electrometallurgical work, possess the advantages that they are less subject to chemical attack, and that they have a far higher conductivity. In addition, they have the further advantage that they can be easily machined and screwed, and therefore assembled in any desired fashion.

As a rule a considerable waste of carbon terminals occurs in electric smelting operations, owing to the butt ends having to be thrown away when they become too short. They can, of course, be used up in other ways, for which, however, a cheaper form of carbon is generally available. The Acheson Company claim that by connecting up the easily-machined graphite terminals as is shown in fig. 1417, that no butt ends are wasted, that the terminals can be continuously fed into the furnace one after er without suspending operations, by simply screwing on another shifting the holder or clamp.

The graphitizing furnaces at Niagara are each of 1000 horse-power capacity. The output of graphite by the International Acheson Graphite Company in 1901 was over 1000 tons. By permission of this company I am able to reproduce here (fig. 1418) a photograph of their furnaces.

Instead of effecting the heating by currents of abnormal density as above, amorphous carbons may be graphitized by passing them continuously through a zone heated by powerful electric arcs, as in the Girard & Street process operated by the Société Le Carbone.

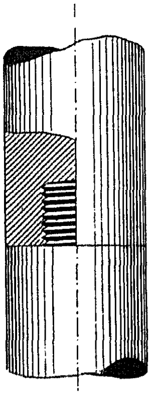


Fig. 1417

Other Electric Furnace Products.—Perhaps the chief of the other electric furnace products is phosphorus. Readman and Parker have shown that phosphates, when electrically heated in presence of carbon and of silica to form a slag, are readily reduced. The phosphorus, being easily volatilized, distils out of the electric furnace, and is condensed, whilst the molten slag can be tapped off. The almost complete replacement of the ordinary method of phosphorus manufacture by the electrical process is due not so much to the high temperature required, as to the application of the heat exactly where it is needed.

Similarly, the manufacture of carbon disulphide, cyanides, &c., is being effected in furnaces which are heated electrically.

Owing to the over-production of calcium carbide in recent years, the plant available for this purpose is being to some extent converted for use in other directions, such as for the production of iron and various iron alloys, and even for the manufacture of glass. The electric distillation of peat has already been referred to.

CHAPTER II

ELECTROLYTIC PROCESSES

General Principles.—The passage of electricity through an electrolytic conductor or electrolyte is conditioned by the motion of ponderable particles, namely the ions. The positively charged ions, or cations, existing in the electrolyte, migrate under the influence of an impressed E.M.F., towards the negative electrode or cathode, while the negatively charged ions, or anions, move towards the positive electrode or anode. Arrived at the respective electrodes they give up their charges of electricity and separate out on the electrodes, or else they react on themselves, on the solvent, or on the electrotrode, with the result that the products we obtain may either be the primary products of electrolysis or be derived from these owing to secondary reactions.

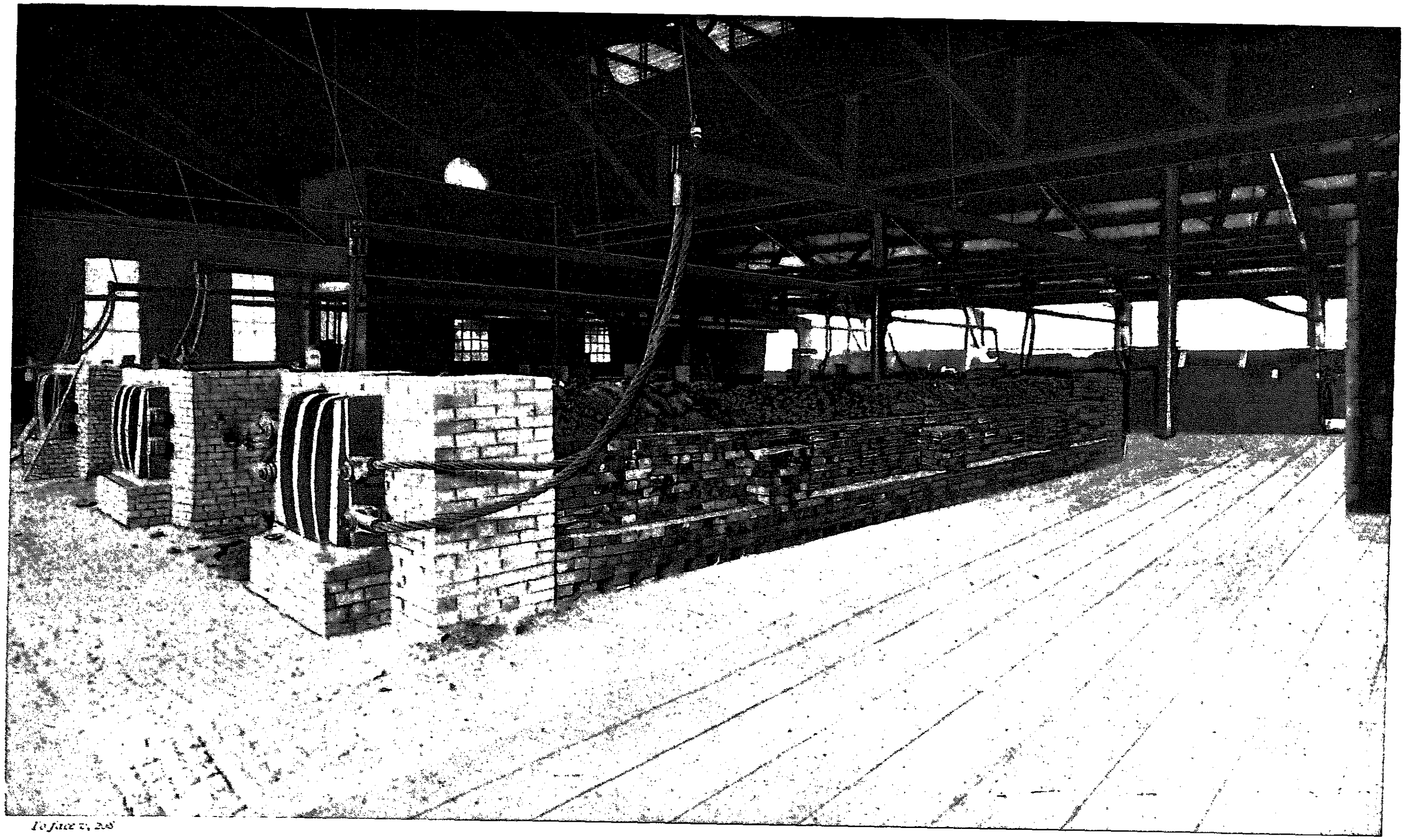
The practical result is the chemical decomposition of the electrolyte, the products, whether primary or secondary, being liberated at the electrodes. The quantitative phenomena of electrolysis are simple and easily accounted for.

When a solution of hydrochloric acid is electrolysed under the proper conditions, equal volumes of hydrogen and chlorine are given off at the electrodes, the weights of the equal volumes being in the proportion of the atomic weights, viz. 1 : 35.45. Similarly, a solution of sodium sulphate yields the secondary products hydrogen and oxygen, the volumes being in the proportion of 2 : 1, and the weights in the ratio of 1 : 8. Fused silver chloride gives 108 parts by weight of silver for 35.45 parts by weight of chlorine gas. In general the weights of the products obtained at the cathode and anode respectively are chemically equivalent to each other, and if the same current is passed through different electrolytes all arranged in series, then chemically equivalent quantities of substances are produced at all the electrodes. Thus, for every one gram of hydrogen set free at the cathode of a cell containing a solution of hydrochloric acid, 108 grams of silver are deposited from a silver nitrate solution, 63.6 grams of copper from a solution of cuprous chloride, 63.6/2 = 31.8 grams of copper from a solution of cupric sulphate, 120/3 grams of antimony from a solution of antimony trichloride, 118.5/4 = 29.6 grams of tin from a solution of stannic chloride, $SnCl_4$, and so on. Compound radicles or ions behave in the same way as the metallic ions. For each gram of hydrogen produced at a cathode, 18 grams of the NH_4 ion (or its products) are liberated at another cathode in series, whilst at the anodes the following ions or the corresponding secondary products are set free: $SO_4/2$ grams from sulphate solutions, $Fe(CN)_6/3$ from ferricyanide solutions, $Fe(CN)_6/4$ from ferrocyanide solutions, and so on, the formula values of all these ions expressed in grams and divided by the valencies being chemically equivalent quantities.

Faraday's laws, expressed concisely and in modern language, state that electricity is only conveyed through electrolytes by the ions, and that equal quantities of electricity are transported with chemically equivalent quantities of the different ions.

If we choose as the electrochemical unit quantity of electricity that quantity of electricity which is transported by one gram-ion of hydrogen, namely, 96540 coulombs, then we know from Faraday's law that the passage of this unit quantity corresponds with the discharge at the electrodes of one gram-ion of every other monovalent ion, half a gram-ion of every bivalent ion, one-third of a gram-ion of every trivalent ion, and so on. Or conversely, the liberation, or deposition, or separation of every monovalent gram-ion requires the passage of 96540 coulombs, every bivalent gram-ion the passage of 2 × 96540 coulombs, every trivalent gram-ion the passage of 3 × 96540 coulombs, &c. Thus the gram-ions may be regarded as vehicles capable of transporting 96540 coulombs, 2 × 96540 coulombs, 3 × 96540 coulombs, &c., according as they are mono-, bi-, or trivalent, &c.

Suppose, for example, we wish to ascertain the maximum or theoretical quantity of sodium chloride which can be decomposed per ampere-hour. A gram-molecule of sodium chloride, NaCl, weighing 23.05 + 35.45 =



To face p. 208

Fig. 1418.—ELECTRIC GRAPHITIZING FURNACES, NIAGARA (International Acheson Graphite Co.)

58.5 grams, yields only the two monovalent ions Na and Cl, and consequently requires for its decomposition the passage of 96540 coulombs, that is to say, 96540 amperes for one second, or 1 ampere for 96540 seconds. One ampere-hour is equal to 1 × 60 × 60 = 3600 ampere-seconds, hence the amount of sodium chloride that can be decomposed per ampere-hour is $\frac{58.5 \times 3600}{96540} = 2.181$ grams.

Again, suppose we wish to find out how much aluminium can be theoretically produced by a current of 1000 amperes acting continuously for a year of, say, 300 days, on a bath of fused alumina, or of alumina dissolved in a suitable flux. In alumina the aluminium is trivalent, hence for the separation of 1 gram-ion, or 27.1 grams, we must pass 3 × 96540 coulombs or ampere-seconds. The maximum quantity of aluminium that can be produced by a current of 1000 amperes in 300 days is therefore—

$$\frac{27.1 \times 1000 \times 300 \times 24 \times 60 \times 60}{3 \times 96540} = 2425 \text{ kilograms.}$$

In this way the current output can readily be calculated. The following table shows, for a few elements, the quantity of each which is separated by the passage of 3600 coulombs, *i.e.* per ampere-hour:—

	Atomic Weight.	Valency.	Yield per Ampere-Hour.
Aluminium	27.1	3	0.3368 gram.
Chlorine	35·45	1	1.3219 „
Copper	63.6	2	1.1858 „
„	63.6	1	2.3716 grams.
Hydrogen	1.01	1	0.03766 gram.
Oxygen	16.00	2	0.2983 „
Potassium	39·15	1	1·4599 „
Silver	107·93	1	4.0246 grams.
Sodium	23·05	1	0.8595 gram.
Zinc	65·4	2	1.2194 „

When secondary reactions take place, the current output of the secondary product is the chemical equivalent of the discharged ion. In the production of caustic soda, *e.g.* by the electrolysis of brine, 1 ampere-hour yields 0.8595 gram sodium, the output of caustic soda equivalent to this being 1.4938 gram NaOH. Similarly, if we look on the formation of chlorates as a process of oxidation, the yield per ampere-hour is the equivalent of the output of oxygen per ampere-hour contained in the above table; or, if we consider the equation, $6\,KOH + 3\,Cl_2 = KClO_3 + 5\,KCl + 3\,H_2O$, it will at once be obvious that for the production of 1 gram-molecule of $KClO_3$ 6 gram-ions of chlorine must be discharged, requiring the passage of 6 × 96540 coulombs. The theoretical output of $KClO_3$ per ampere-hour is thus

$$\frac{122.6 \times 3600}{6 \times 96540} = 0.762 \text{ gram.}$$

In actual practice the current efficiency is in general lower than the theoretical, depending on various other conditions, on the construction of the plant, &c. For example, the products, separated at the electrodes, may, if no provision is made to prevent it, recombine or react with each other, or undesirable secondary reactions may occur, or the current may be transported more or less by the products formed instead of by the substance required to be electrolysed, and so on.

We have already seen that there is a definite relationship between electrical energy and other forms of energy, such as mechanical energy, heat energy, &c. 1 volt-coulomb = 10^7 erg = 0.102 kilogram-metre = 0.24 gram-calorie. Each form of energy can be represented as the product of two factors, one of these being of the nature of a capacity or quantity, whilst the other is an intensity or potential. Electrical energy, *e.g.*, is the product of the quantity expressed in coulombs, say, and of the potential expressed in volts. A similar relationship must exist between electric energy and chemical energy, but it is unfortunately the case that at present we know next to nothing of chemical energy or of its factors. If, however, we measure the change in chemical energy by the heat evolved or absorbed during the chemical reaction, and if we further assume that the whole of this heat can be converted into electrical energy and made available as an electrical current, then we would simply have the relationship—

Electrical energy (expressed in calories) = Heat of the reaction.

The exact measure of the electrical energy in volt-coulombs is $E \times 96540$ for a monovalent, or $E \times n \times 96540$ for an n-valent gram-molecule; or, since 1 volt-coulomb = 0.24 gram-calorie, the exact measure of the electrical energy in gram-calories is $E \times n \times 96540 \times 0.24$. If the corresponding heat of the reaction is Q gram-calories, then the relation would be—

$$E \times n \times 96540 \times 0.24 = Q,$$

$$\text{or} \qquad E = \frac{Q}{n \times 96540 \times 0.24}.$$

If this relationship, usually known as Thomson's theorem, were generally applicable, we could easily calculate the electromotive force of any primary battery or the minimum electromotive force necessary to reverse a chemical reaction or decompose a compound. At one time it was thought to be strictly true, and as a matter of fact it is very nearly true in a few particular cases, more especially in the case of the well-known Daniell battery. The reaction which takes place in the Daniell cell is $Zn + CuSO_4 = ZnSO_4 + Cu + 50130$ gram-calories. Applying the above equation we have—

$$E = \frac{50130}{2 \times 96540 \times 0.24} = 1.08 \text{ volt},$$

which is very nearly the average E.M.F. of the cell. More recent investigations—which so far, however, are only applicable to strictly reversible processes, such as the Daniell cell—have shown that in general the electrical energy is not equivalent to the heat of the reaction. If we represent by EC the electrical energy, then the correct relationship for such cells is—

$$EC = Q + T\frac{dE}{dT},$$

where T is the absolute temperature, and dE/dT the temperature coefficient, or the change in E.M.F. for one degree Centigrade. dE/dT may have either a positive or a negative value, and only when the expression $T\,dE/dT$ has the value zero, or when it can be shown that heat is neither given out to nor absorbed from the environs, is it permissible to equate the electrical energy and the heat of the reaction.

Sometimes, however, when only a rough approximation to the E.M.F. required for decomposition is wanted, no attention is paid either to the reversibility of the cell or to the temperature coefficient. In such cases little more than the order of magnitude of E can with certainty be deduced. The value so found is in general neither the minimum electromotive force necessary, nor yet, properly speaking, the theoretical value. If it is desired to estimate the least amount of energy to be supplied by a dynamo for any process, the only satisfactory way of doing so, as far as we know at present, is to determine experimentally the decomposition potential of the electrolyte, or the lowest E.M.F. required to produce a sensible current, and then to combine this with the theoretical current output calculated from Faraday's law. In actual practice a higher E.M.F. will, of course, be required, for economic reasons among others, while in general the current output will be less than the theoretical. The ratio of the energy actually consumed to that calculated on the above basis is the nearest approach we can make to the energy efficiency of the process.

Electrolytic Refining and Treatment of Copper and other Metals.—Electrolytic copper refining is now a very large business. The annual production of crude copper is not much short of half a million tons, and of this about half is electrolytically refined. The crude copper which is refined usually contains about 95 per cent or more of metallic copper, the remainder being impurities. The object to be attained in copper refining is twofold. In the first place, it is desirable to obtain purer copper. The influence of small quantities of impurities on the properties of a metal or other substance is, as is well known, very considerable. The electrical conductivity of pure copper, *e.g.*, is much greater than that of copper containing a few per cent of impurities; and since a large proportion—over 60 per cent—of the annual output of copper is used for electrical conductors, it is important that this should be as pure as possible. It has been estimated that the annual saving effected by using pure electrolytic copper exceeds one and a half million pounds sterling. In the second place, the impurities existing in crude copper contain a very considerable proportion of valuable metals, chiefly silver and gold, which in the process of refining are obtained in a concentrated condition suitable for further working up for the recovery of these values.

Although there are several modifications of the refining process, the general principles are usually the same. Two copper electrodes submerged in a solution of copper sulphate constitutes what is known as a non-polarizable cell. Within all electrolytes Ohm's law is valid, but in most cells during electrolysis alterations take place at the electrodes whereby an opposing E.M.F. or polarization is set up. This counter E.M.F. may be due to a change in the surface of the electrodes, to the deposition of another metal, to the separation or occlusion of gas, or to a change in the concentration of the electrolyte, &c. In a strictly non-polarizable cell, which would obviously be extremely difficult of realization, all these disturbing influences would require to be absent. The cell we have just referred to, however, is an approach to a strictly non-polarizable cell, inasmuch as an extremely minute impressed E.M.F. produces a current in accordance with Ohm's law. During electrolysis the copper ions give up their charges of electricity and separate out on the cathode in the neutral state, whilst the sulphions (SO_4'') similarly give up their charges at the anode. Under suitable conditions of current density they attack the anode, so that as much copper goes into solution there as is deposited on the cathode. The whole process, therefore, consists in the dissolution of the anode and in the precipitation of copper on the cathode, or in the transference of copper from anode to cathode.

In actually carrying out the process of copper refining a number of anodes, consisting of cast plates of the crude copper, are arranged in a bath containing 15 per cent crystallized copper sulphate and 5 per cent sulphuric acid. Between each of the anodes, and also beyond the extreme anodes, are suspended thin sheets of pure copper, which act as cathodes. All the anodes and all the cathodes are connected in parallel with, or are suspended from, two insulated omnibus-bars supplying the current. The number of plates in each bath, and their dimensions, vary in different works. The vats are usually constructed of pitch-pine lined with lead. Between each plate it is usual to allow a distance of about 2 inches, and provision has also to be made for the accumulation and collection of the mud which falls from the anodes.

Theoretically, an insignificant E.M.F. is required per bath, and consequently a very small quantity of electrical energy should suffice to transfer an enormous quantity of copper from the anode to the cathode. In order, however, to keep the consumption of electrical energy at a minimum, it would be necessary to make the number and cross-sections of the baths a maximum to obtain a specified output of refined copper in a given time. Electrical energy has to be paid for, and must therefore be kept as low as possible. On the other hand, an indefinite extension of plant and works has also to be paid for. Besides an increase of ground rent, and of interest and depreciation on capital expended for plant, there is the interest on the stock of copper held both in the form of crude copper for anodes and of copper in the electrolyte. It is undoubtedly true that in a great many electrochemical industries the cost of the electrical energy is one of the most serious items in the cost of production. This, however, is not the case to the same extent in the electrolytic refining of copper, where it is more economical to increase the expenditure for electrical energy in order to increase the output of copper, and so diminish the charges for the stock of copper held, &c. In practice a compromise must be effected between the outlays in different directions, in order to achieve the most economical results

It is customary to employ an E.M.F. of about 0.3–0.7 volt per cell, and a current density of about 10 amperes per square foot of cathode surface. Owing to the low voltage required, therefore, it goes without saying that a number of such cells must be arranged in series. The omnibus-bar connected with the anodes in one cell will accordingly be placed in communication with the cathodes in the next cell, and so on.

The current density and the circulation of the electrolyte are most important practical considerations. In order to obtain a good coherent deposit of copper the current density, *i.e.* the number of amperes per unit of surface, must not greatly exceed the figure already stated for a slow circulation of the electrolyte. To obtain a deposit of copper at all there must be a sufficiency of copper ions at the cathode, otherwise the separation of hydrogen will cause irregularities in the resistance and in the deposition of copper on the cathode. As soon as the copper deposited at any one point of the cathode projects farther towards the anode, the current density will become greater at this point, and the tendency will be for it to grow outwards at the expense of those portions of the general surface more remote from the anode.

Again, unless the electrolyte is stirred or agitated, the concentration of copper sulphate in the neighbourhood of the anode may increase faster than diffusion can equalize the concentration of the electrolyte, in which case copper sulphate may crystallize out on and protect the anode from further action. Provision has, therefore, to be made for the constant circulation or stirring of the electrolyte. This may be effected either by arranging the cells on a terrace and allowing the electrolyte to flow by gravity through the series into a sump, whence it is pumped or forced back to an overhead tank, or by keeping the electrolyte in each cell in motion by means of a suitable contrivance.

From time to time the electrolyte must either be purified or replaced by fresh solution. The impurities contained in the anode are of two kinds. Some are insoluble in the electrolyte, and are therefore either left adhering to the anode, or else fall to the bottom as a fine mud, whilst others are soluble in the solution. Working under proper conditions with a low voltage, and so long as a sufficiency of copper is present, the latter are not deposited on the cathode; but as they gradually accumulate they displace the copper and render the solution less and less pure, so that finally they must be removed.

The amount of electrical energy required to refine copper is easily computed. For 1000 kilograms or a metric ton (2204 lbs.) of copper it is calculated as follows:—The atomic weight of copper is 63.6, and since it is divalent the deposition of 63.6 grams requires the passage of 2×96540 coulombs or ampere-seconds; 1000 kilograms, therefore, necessitate the passage of $\frac{2 \times 96540 \times 1000000}{63.6}$ ampere-seconds, provided the current efficiency is 100 per cent. By multiplying by 0.5 volt, the E.M.F. supposed to be employed, we get the energy measured in watt-seconds. Hence the energy required in kilowatt-hours is $\frac{2 \times 96540 \times 1000000 \times 0.5}{63.6 \times 1000 \times 3600}$ = 421.7 kilowatt-hours per metric ton. Assuming that with a medium-sized steam plant working continuously the kilowatt-hour costs ½*d.*, then the cost of the energy required to refine a metric ton would be about 17*s.* 6*d.*, and correspondingly less if a cheaper source of energy is available. The other charges in good practice bring the total cost of refining up to a little over £3 if the value of the silver and gold is not taken into account.

Besides the above parallel arrangement of the anodes and cathodes in the vat, other systems are possible. The chief objection to this system is that unless the electrical contacts are very good, the current density on the different cathodes is apt to vary considerably owing to the low E.M.F. employed. To overcome this the plates are sometimes, but not often, arranged in series in the vats. In Hayden's process no cathodes are employed, and the plates nearly or completely fill the cross-section of the vats, so that the lead lining must be dispensed with. The leads are attached to the end-plates, and each intermediate plate acts as a double-pole electrode, *i.e.* copper is dissolved off one face of the plate, while pure electrolytic copper is deposited on the other. When nearly all the copper has been thus dissolved away the pure copper is stripped off.

The rough plates of pure electrolytic copper as they are removed from the vat are not of much use as they stand. Before they can be rolled into sheets, or drawn into wire, they have usually to be first melted down into ingots. This not only increases the cost, but also causes the introduction of more or less impurity. Several special processes have been introduced to obviate this difficulty.

In the Elmore process the cathode is a revolving cylinder or mandrel. The copper deposited on this is kept smooth and bright by means of an agate burnisher or tapping arrangement, which automatically traverses the copper tube from end to end, and thus prevents the formation of excrescences and unequal current density. When a sufficiently thick deposit of copper has been attained the cylinder is removed, and the copper tube expanded off. In order to prevent the tube from sticking to the mandrel, the latter must be covered with a thin film of grease or graphite, &c. Such tubes of almost pure copper have a tensile strength of about 20 tons per square inch. By means of a longitudinal cut wide tubes can then be folded out into sheets, or if wire is desired, the tubes are cut spirally, and the spiral of square cross-section drawn into wire of the required size.

It has already been pointed out that in the ordinary process of copper refining the current density must not much exceed 10 amperes per square foot if the electrolyte is kept in slow circulation.

In the Cowper-Coles centrifugal process, however, smooth non-laminated tubes having a diameter of 12 inches can be produced with a current density of 200 amperes per square foot by causing the mandrels to rotate rapidly in the electrolyte. Hydrogen or other impurities, which might accumulate on the surface and prevent the formation of a smooth uniform deposit, are thus thrown off. In other processes a high current density can also be employed, as, *e.g.*, when jets of the electrolyte are projected against the cathode.

Radically different from the process of copper refining are some processes which have been proposed for the extraction of copper from its ores. A higher E.M.F. is required, and consequently the consumption of electrical energy is greater. In the Siemens and Halske process, a solution containing copper sulphate and ferrous sulphate, obtained by lixiviating cuprous sulphide with ferric sulphate solution, is passed through the cathode compartment of a cell divided into two by a porous partition. After the copper is deposited, the resulting liquor, consisting substantially of ferrous sulphate solution, is then passed through the anode compartment containing a carbon anode, where it is oxidized to ferric sulphate, which is returned to the lixiviating vats to extract a fresh portion of copper. The equation, $Cu_2 S + 2 Fe_2 (S O_4)_3 = 2 Cu S O_4 + 4 Fe S O_4 + S$, represents the action in the leaching process, whilst $2 Cu S O_4 + 4 Fe S O_4 = 2 Cu + 2 Fe_2 (S O_4)_3$, shows the action which takes place during electrolysis. Another ingenious process is that of Hoepfner, but so far as is known at present neither seems to be in successful operation on a large scale. Before quitting this part of the subject it should be noticed that the mere deposition of a metal from a solution, or its transference from a moderately pure anode to a cathode, does not necessarily imply that the metal so deposited is in any way a purer or refined metal. On the electrolysis of any solution with unattackable electrodes, the electromotive force of polarization e is made up of the E.M.F. e_1 required to discharge the cation at the cathode, and of the E.M.F. e_2 to separate the anion at the anode, thus $e = e_1 + e_2$. If, however, several different cations and anions are present in solution, electrolysis will commence as soon as the E.M.F. is raised just sufficiently high enough to discharge any one of the cations and any one of the anions. Different metals require different electromotive forces to discharge them, and it is owing to this fact that they can be separated by electrolysis. So long as the E.M.F. exceeds that required for one metal, and still remains below that required for another, the first metal will be deposited in the pure state, and when it is all thrown down the current will cease. On raising the E.M.F. the next metal will be deposited, and so on. If the higher E.M.F. is used at the start, then in some cases both metals are deposited simultaneously, as in the process of electro-brassing.

The preceding very brief outline of the electrometallurgy of copper must here suffice to indicate the general principles underlying the electrolytic refining and recovery of metals in aqueous solutions.

Moderately pure gold and silver are refined by processes analogous to that of copper refining. Gold is recovered from its solution in dilute potassium cyanide by the Siemens and Halske process. In this process the gold solution is electrolysed between thin sheet-lead cathodes and iron or graphite anodes, with a current density of about one-twentieth of an ampere per square foot, and when a sufficient quantity of the gold has been deposited on the cathodes they are removed and cupelled. The amount of electrical energy required is very small, and the cost of the electrolytic recovery is about the same as the precipitation of the gold by means of zinc. It offers the advantage, however, that much more dilute cyanide solutions can be successfully treated.

Alloys of gold and silver are separated or parted electrolytically. In this case anodes are cast of the alloy, whilst thin sheets of silver form the cathodes. The electrolyte used is a solution of silver nitrate, and the current density employed is about three times as high as in copper refining.

Zinc extracted from the ores in the form of a chloride solution is, after purification, deposited on suitable cathodes. The chief difficulty connected with the recovery of zinc is due to the fact that unless special precautions are taken it is apt to separate out in the spongy condition instead of in a coherent reguline mass. At the works of Brunner, Mond, & Co., almost pure electrolytic zinc is produced as a by-product in recovering the chlorine from waste calcium chloride liquors.

Many processes have been proposed for the electrolytic stripping of tin from waste tinned iron or scrap, which usually contains about 3 per cent of tin. Assuming that an adequate supply of the waste material is available, the tin can be removed by causing the material to act as anode in a bath of dilute acid, or preferably of caustic soda. The tin which is dissolved off the anode is deposited on a suitable cathode, but generally in a spongy condition which renders further treatment necessary. The Goldschmidt works at Essen are stated to treat annually 10,000 tons of scrap, which after the removal of the tin forms a good raw material for steel works. Other works in Germany treat another 20,000 tons of scrap.

Processes have also been devised for refining lead, for the extraction of lead and antimony from their ores, &c. The electrometallurgy of nickel is of considerable interest and importance, but so far no authoritative complete account of any commercial undertaking has been made public.

Electrodeposition of Thin Films of Metals.—What may now be regarded as the art of electroplating and electrotyping is perhaps the oldest electrochemical industry. It was practised nearly one hundred years ago with the help of primary batteries, and long before the modern efficient dynamo was invented. Although individual establishments may not require the expenditure of very large amounts of electrical energy, yet in the aggregate the industry is a very large one. It is, of course, impossible to enter into any details here with respect to the technique of electrotyping or electroplating.

Up to the present we have been concerned more with the application of heavy electrical currents and with the production of thick deposits of metal either in the process of refining or of manufacturing metals in aqueous solutions. We have now to consider the deposition of comparatively thin films of metals, such as are required in these processes.

In electrotyping, the object to be attained is as a rule the reproduction in copper of an object either in relief or intaglio. To accomplish this a mould or cast of the object to be copied is first taken by means of a suitable medium or composition. If the latter should not happen to be an electrical conductor, the surface to be reproduced must be rendered conducting by the application of graphite, or in some other way. Assuming that an exact copy in some plastic material has been secured, and that the surface has been rendered conducting, and that further the surface is not so absolutely clean that the electrodeposited copper will adhere too firmly to it, then it is only necessary to make such a cast the cathode during the electrolysis of a copper sulphate solution. In order to maintain the electrolyte at the initial concentration, and also diminish the expenditure of energy, an anode consisting of pure copper should be employed. When a film of sufficient thickness has been deposited, observing the general conditions necessary for the refining of copper, it is then detached from the mould and backed with fusible metal or otherwise as required.

Electroplating, on the other hand, differs essentially from electrotyping, inasmuch as the film of electrodeposited metal is intended to adhere as completely and as firmly as possible to the object being treated. It is therefore necessary that the surface should be rendered smooth, bright, and, in particular, clean, from the chemical point of view. This can, of course, be accomplished in various ways, which do not require to be specified. After acting as cathode in a suitable bath until a sufficiently thick film is deposited, the plated object is removed, washed, and burnished mechanically when this is possible. Gold, silver, and copper, for example, are soft and easily burnished. Nickel, on the other hand, is extremely hard and difficult to burnish, so that the greatest care ought to be taken in obtaining a smooth, bright surface before electroplating.

As has already been indicated, the character of the deposited metal is influenced to a great extent by the current density employed, and by the rate at which the electrolyte is stirred or circulated. Since the latter influences the character of the deposit by maintaining a constant composition of the electrolyte in the neighbourhood of the cathode, it will readily be understood that any variation of the concentration of the electrolyte will affect both the deposit and the permissible current density. Other factors, such as the acidity or alkalinity of the electrolyte, have naturally also to be considered in individual cases. If basic salts are objectionable, for example, the addition of acid would not only prevent their formation, but also alter the resistance of the electrolyte. The addition of various organic and other substances, often in very minute quantity, has sometimes a marked and hitherto unexplained influence on the character of the precipitated metal.[1] Bunsen also observed, many years ago, that metals deposited as the result of a secondary reaction have, as a rule, a more uniform and beautiful surface; and it would seem that the favourite baths now in use are solutions of complex salts, from which the metals thrown down are not the primary products of electrolysis. An example may make this clear. In a simple solution of silver nitrate the cations are Ag. During electrolysis these are discharged and separate out at the cathode, generally in a more or less granular or crystalline form. Crystals having once formed, the successive quantities of deposited silver will tend to grow on and increase these. On the other hand, the cations, in the case of

[1] Alcohol, urea, hydroxylamine, pyrogallol, &c., probably prevent the formation of oxides or basic salts.

solutions of complex salts, are not the metals which are ultimately deposited. Potassium argentocyanide, $K\,Ag\,(C\,N)_2$, as Hittorf has shown, yields the positive ion $\overset{+}{K}$, which migrates towards and is discharged at the cathode, whilst the complex negative ion $\overset{-}{Ag\,(C\,N)_2}$ travels to the anode, where it also gives up its charge. The silver actually deposited on the cathode is therefore the result of a secondary reaction, which may be represented thus: $K + K\,Ag\,(C\,N)_2 = Ag + 2\,K\,C\,N$. In this case the precipitation of silver will take place at the points where the potassium ions are discharged, and since these have no tendency to separate out on any particular existing particles of silver, the deposit will be more smooth and uniform. Other complex salts, such as potassium gold cyanide, ammonium nickel sulphate, &c., behave in the same way.

The temperature of the bath is another important factor. Increase of temperature in the case of a primary process influences the electrolysis in so far as it reduces the resistance of the electrolyte. Secondary processes, on the other hand, are purely chemical, and as such are affected to a much greater extent by rise of temperature. The deposition of a metal from a complex salt, being a secondary and therefore a purely chemical process, is likely to be promoted by warming the bath, as, for example, in gold and nickel plating with the above-mentioned salts. Nearly all the metals, with the exception of the alkali metals, and magnesium, aluminium, &c., can be electrodeposited from aqueous solutions; but for further details or recipes the reader must consult special works.

Before leaving this subject it should be noted that the object of electroplating generally is either to embellish or protect the article. In the case of protection the metallic film should not only cover the object completely, but it should also act as a true protecting coating. Gold or platinum, *e.g.*, are not generally acted on by corrosives, and therefore act as efficient but expensive protections. The less noble metals behave more or less efficiently according to circumstances, and, paradoxical as it may appear, some exert an excellent protective action because they are more or less easily corroded or acted on. A case in point is zinc. Iron or steel coated with zinc may or may not be efficiently protected from corrosion according to circumstances. Even although the film of zinc was imperfect, steel would be effectively protected from corrosion provided that it was submerged in the corroding medium and that any considerable quantity of zinc was left at all.

Both steel and zinc, for example, taken separately, would be attacked by a dilute acid. In contact, however, they form a galvanic couple, and zinc being the more positive metal would be attacked and dissolved in preference to the steel, on the surface of which only hydrogen would be evolved; so that as long as zinc is present the steel is protected from corrosion. For this reason it is now customary to protect the boiler-tubes, boilers, or hulls of vessels, &c., with zinc. This can, of course, be done by dipping the parts in molten zinc according to the older process of galvanizing, but the tensile strength of steel of high breaking load is thereby diminished. Electrolytic zincing or cold galvanizing, on the other hand, can be done at the ordinary temperatures, and it is claimed that the Cowper-Cowles regenerative process of electrolytic zincing is not only cheaper, but that the deposit of zinc is more adhesive. The tubes, boilers, and hulls of between forty and fifty torpedo-boats, destroyers, &c., have been wholly or partially coated by zinc by this process. Galvanized articles, whether by the hot or cold process, exposed merely to the atmosphere and not submerged in an electrolyte, are, of course, not protected to the same extent.

Electrolytic Alkali and Chlorine Industry.—Starting with sodium chloride the Leblanc and ammonia-soda processes convert the sodium of the salt by a series of operations into one or other of the following compounds:—Anhydrous sodium carbonate (soda-ash), sodium carbonate crystals, sodium bicarbonate or caustic soda. In the former process the chlorine is obtained in the first instance as hydrochloric acid, which can either be utilized as such or converted into bleaching-powder, &c., whilst in the latter process the chlorine is more or less run to waste in the form of calcium chloride liquors. In the electrolytic alkali industry, which is now entering into serious competition with the older chemical processes, the salt, either in the form of brine or in the fused condition, is decomposed by electrolysis into the primary products sodium and chlorine at the cathode and anode respectively. The quantity of electrical energy required for the decomposition under different conditions is known. This, however, is only the beginning of the problem. The chlorine discharged at the anode is given off more or less completely as a gas, and is thus removed from the sphere of action, but not before a small quantity has undergone secondary reactions which leave their mark behind either in the shortened life of the carbon anodes, or it may be in a reduced current efficiency or in the quality of the liberated chlorine, &c. The difficulties in the way of recovering the whole of the discharged sodium are also considerable. When the sodium cannot act on the cathode or on the electrolyte it should, of course, be obtained in the metallic state. Many attempts have been made to obtain the metal in this way, but the difficulties connected with the construction of plant on the large scale have been so great that no commercial success can be recorded. Grabau & Borchers have, however, constructed apparatus which can be employed on a small scale for the production of sodium from fused salt.

In the existing electrolytic alkali processes, therefore, the sodium is either allowed to react on water, whereby caustic soda and hydrogen gas are produced, or it is absorbed by the cathode and removed from the sphere of action. In any case the difficulty consists in removing the sodium or its secondary products from the undecomposed salt, and although the number of ways in which this can be done is considerable only a few of these have so far proved commercially successful.

The aim of this brief account is to indicate the main principles underlying the construction and method of working of the chief types of electrolytic cell, and it may as well be stated at the outset that the full practical details of most of these have never been published and are carefully guarded as trade secrets.

In most processes a salt solution or brine little short of saturation is submitted to electrolysis. Should such a solution contain objectionable impurities these must first of all be removed by chemical methods. The fundamental reaction during the electrolysis is simple, and may be represented by the equation $NaCl = Na + Cl$. When the sodium acts on the solvent we have $Na + H_2O = NaOH + H$, or, combining both equations,

$$2NaCl + 2H_2O = \underbrace{2NaOH + H_2}_{\text{Cathode.}} + \underset{\text{Anode.}}{Cl_2}.$$

These apparently very simple reactions are complicated by at least seven or eight secondary reactions, and before they could be realized on a practical scale very large sums of money have had to be expended on experimental trials.

The main problem of the separation of the cathode and anode products may be considered under gravity processes, diaphragm processes, and processes depending on the intervention of a mercury cathode which dissolves the sodium, which can then be made to react with water outside the cell proper.

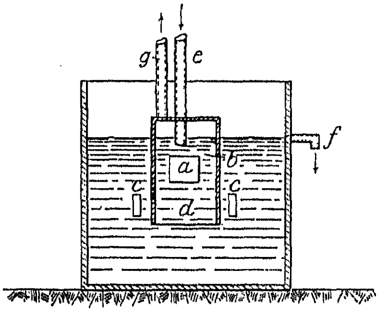


Fig. 1419.—Gravity Process Cell

Gravity Processes.—Several gravity processes have been proposed, but the general principle will be understood from fig. 1419, which represents diagrammatically a cell patented by the Oesterr. Verein für Chem. und Metallurg. Produktion in Aussig. The anode *a*, made up of a number of carbons, is placed within an inverted earthenware bell *b*, which is suspended inside an iron containing vessel. The ring-shaped iron cathode *c* encircles the bell at a little distance from its mouth. The caustic soda formed at *c*, being specifically heavier than the brine, accumulates in the bottom of the tank, whilst the chlorine liberated at *a* escapes through the pipe *g*. During electrolysis the neutral interface at *d* between the causticized liquor in the lower part of the bell and the brine saturated with chlorine gas in the upper part, gradually becomes displaced, but by feeding in fresh brine through the pipe *e* at the proper rate, and by permitting the causticized liquor to overflow at *f*, the position of the interface is maintained at a constant level, and secondary reactions are prevented to a large extent. The hydrogen given off at the cathode escapes into the air or may be collected.

This cell, or a modification of it, is understood to be in operation at the above and some other Continental works, probably for the electrolysis of potassium chloride, and giving a current efficiency of about 80 per cent. A similar gravity process, viz. that of Holland & Richardson, has been tried by the Electrochemical Company on a large scale at St. Helen's in Lancashire, the electrolyte being brine, but the current efficiency only reached about 66 per cent, the cathode current density being about 10 amperes per square foot and the voltage about 5 volts per cell. The merits of the gravity process are its simplicity and the absence of a diaphragm, whilst its disadvantages are the comparatively low current efficiency and the contamination of the caustic liquor by undecomposed salt, which can, however, be separated during evaporation.

Diaphragm Processes.—In these the cell is divided into two compartments, viz. an anode and a cathode compartment, by means of a porous diaphragm, which will permit of the passage of the electric current, but at the same time prevent the intermingling of the products. Diaphragms may be made of all kinds of materials, *e.g.* porous earthenware, specially treated cement, asbestos mixed with other materials not chemically acted on and suitably bound together. Besides these, burnt magnesia and powdered limestone, soap, &c., have been proposed. The chief objection to the introduction of a diaphragm is the increase of electrical resistance which it occasions, and consequently the increased consumption of electrical energy. Further, if the life of a diaphragm is short, its renewal means additional expense, directly in so far as the material itself and labour are concerned, and indirectly by the stoppage of a process which, if it is to be conducted economically, must be continuous. Diaphragms must, of course, possess the required mechanical strength, but they need not necessarily give way as a whole before they are rendered useless. The appearance of little more than pin-holes may be sufficient to condemn them.

With regard to electrodes, iron or copper in some shape or form is almost universally adopted for cathodes. Nickel and copper oxide have sometimes been suggested. The material of which anodes may be constructed is strictly limited. Carbon in some form, and platinum, or preferably an alloy of platinum with 10 per cent of iridium, are usually employed; but ferro-silicon, magnetite, and phosphor-chrome have been proposed. Although platinum is very expensive, it can now be obtained in the shape of very fine foil and gauze. Artificial amorphous carbon and gas carbon blocks, cast into lead if necessary, are employed to a considerable extent; but graphite electrodes, produced in the electric furnace by the Acheson and by the Girard & Street processes, possess many advantages as regards electrical conductivity, resistance to chemical action and mechanical disintegration, and ease of assembly, so that they are now adopted to a very large extent. Cast-iron, lead, slate, ebonite, earthenware, cement, &c., can be employed for different parts of the cell, depending, of course, on whether the material is in contact with alkali or chlorine.

In spite of the various defects that have been urged against diaphragm processes, some of them appear to meet with a fair share of success. Cement diaphragms, *e.g.*, have been used for years on the Continent for the electrolysis of potassium chloride solutions. The advantage of electrolysing potassium chloride instead of sodium chloride lies in the fact that a stronger solution of the former with a lower resistance can be obtained, that the current output per ampere-hour is greater, and that the money value of caustic potash is higher.

The cement diaphragms employed on the Continent are probably prepared by incorporating with the cement a salt, or sulphur or other material, which is subsequently dissolved out after the cement has set, but the exact procedure is kept secret.

The following figures represent some typical forms of cell in which a diaphragm is employed. Fig. 1420 illustrates the principles underlying the construction of the Le Sueur cell, a modification of which is in operation at Rumford Falls, in America. The mouth of the inverted bell B, of slate or other suitable material, is closed by an asbestos or vegetable-parchment diaphragm r^2, and immediately underneath this is the wire-cloth cathode t^1. In the anode chamber are suspended the carbon or platinum anodes b^3. The whole is slightly tilted to allow of the escape of the hydrogen from the cathode. Further details of a later construction of the cell are contained in the *Jour. Am. Chem. Soc.*, 20, 11, 868. Owing to the high E.M.F. required, viz. 6.5 volts or more, such a cell could only be operated where electrical energy is cheap.

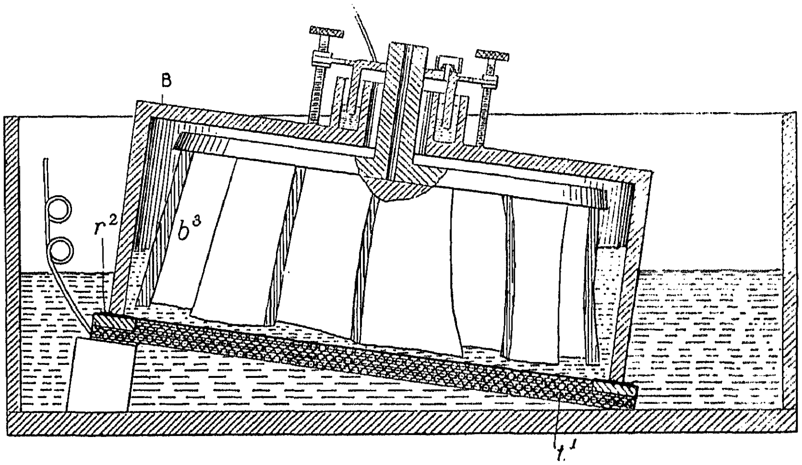


Fig. 1420.—Diaphragm Process: Le Sueur Cell

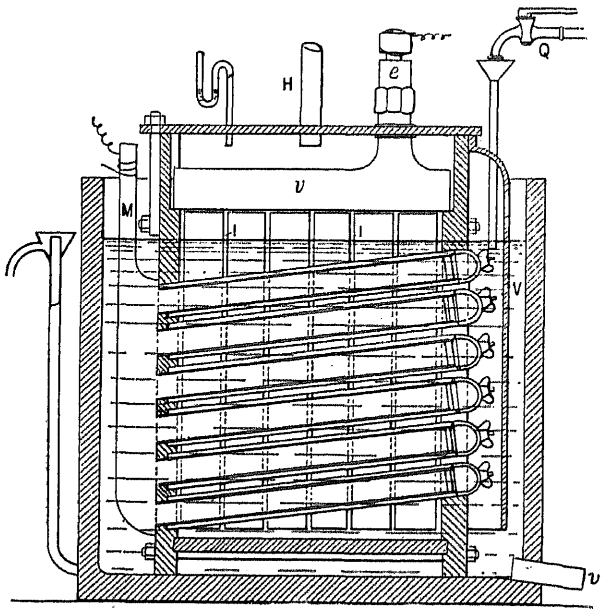


Fig. 1421.—Diaphragm Process: Outhenin-Chalandre Cell

Fig. 1421 illustrates the principles of the Outhenin-Chalandre cell, extensively employed in France and Switzerland. The anode chamber supplied with brine contains the carbon anodes 1 connected with the positive terminal *e*. Each vertical set of these is separated by a slightly-inclined set of porous earthenware or other diaphragm tubes, each of which contains a strip of sheet-iron attached to the main cathode terminal M. A feature of the cell is that the whole anode chamber can be raised for the purpose of cleaning, examining, or repairing. During electrolysis the chlorine is evolved at the carbons, and passes off through the pipe H. The cathode compartment is filled at the commencement with water or a dilute solution of soda, which gradually becomes stronger owing to the separation of caustic at the cathodes inside the porous tubes. The hydrogen, which is given off from the cathodes, passes up the inclined tubes and escapes into the air or is collected in the chamber V. As soon as a sufficiently strong solution has been obtained, it is run off at *v* and replaced by water from the tap Q. In more recent modifications of this and of the Le Sueur cell a special point is made of keeping the brine always slightly acid by the introduction of hydrochloric acid. The object in so doing is to decompose and prevent the accumulation of hypochlorite or hypochlorous acid produced by the intermingling of the cathode and anode products, thereby prolonging the life of the anodes and ensuring the production of purer chlorine gas.

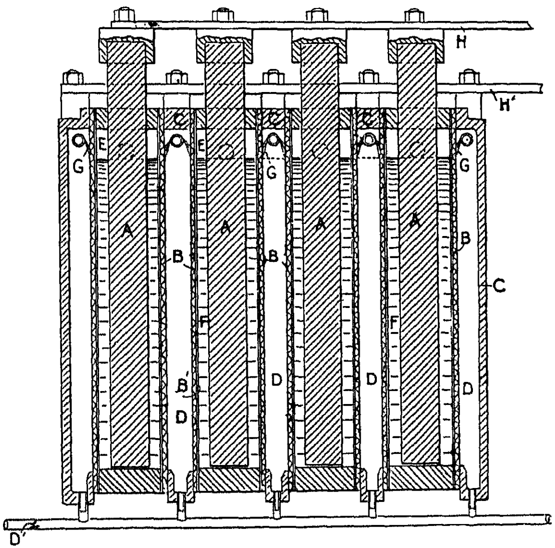


Fig. 1422.—Diaphragm Process: Hargreaves & Bird Cell

The third and last type of diaphragm cell which can be referred to here is that of Hargreaves & Bird. A diagrammatic vertical section is shown in fig. 1422. The carbon anodes A connected with the positive conductor H are immersed in the brine to be electrolysed. This brine is retained by composite cell walls, consisting of a special diaphragm material B′ formed on wire-cloth cathodes B, placed in communication with the negative pole H′. In the diaphragm cells previously alluded to it will have been noticed that the cathodes were plunged in water or dilute alkali solution, which subsequently became more concentrated owing to the production of caustic soda during the electrolysis. A special feature of the Hargreaves & Bird cell is that the composite cathode diaphragm is more or less impervious to the salt solution as long as the cell is not in operation. During the passage of the current, chlorine is given off at the anode as usual, and is led away, whilst the sodium ions pass through the diaphragm B′ and are discharged on the cathode B. Here they react with a fine spray of water or steam, introduced at G into the cathode space D, producing hydrogen and caustic soda, which is at the same time washed off the cathodes in a more or less concentrated condition. In practice, however, the Electrolytic Alkali Company of Middlewich, who are now working the process, find it preferable to introduce carbon dioxide with the steam, and so produce a concentrated solution of sodium carbonate, which runs off at D′, and is crystallized, after further evaporation if necessary. Provision is made for circulating the brine, which, after being impoverished and freed from dissolved chlorine if necessary, is run back to the brine wells to take up a fresh portion of sodium chloride before being returned to the process. The peculiar cathode diaphragms are stated to be made from a mixture of asbestos, Portland cement, and silicate of soda, which is felted

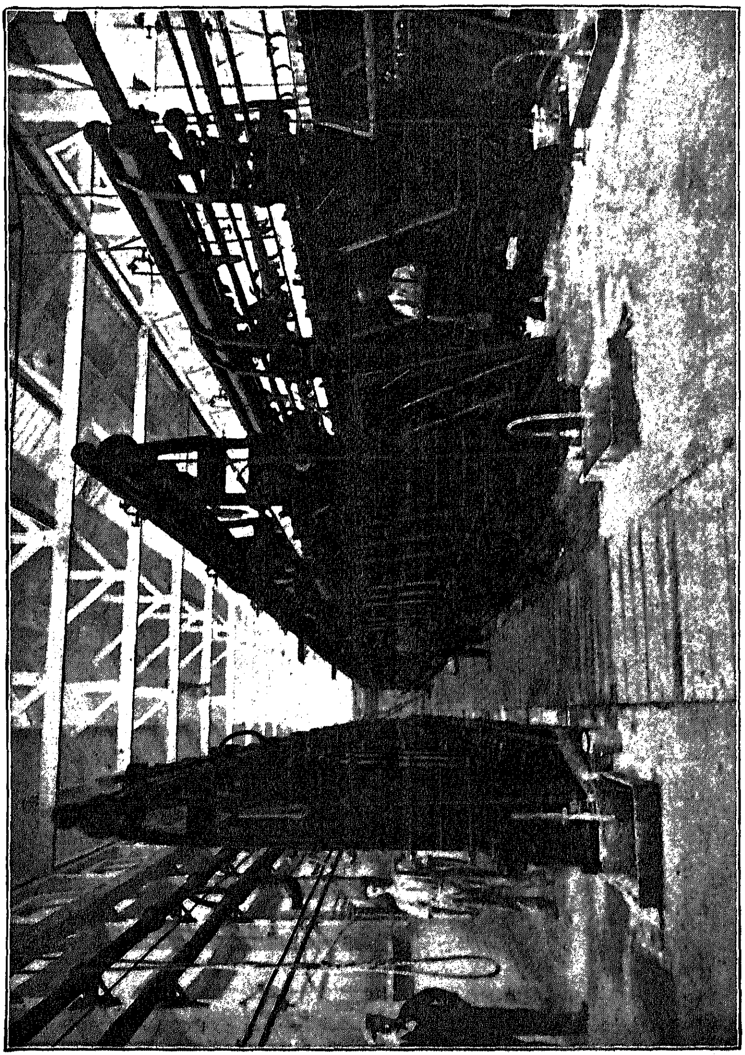


Fig. 1423.—Cell-Room at the Electrolyte Alkali Company's Works

on to the copper-gauze cathode. These diaphragms and the sides of the cell are so bolted up that the removal of a diaphragm (possibly once every two or three months) will occupy a minimum of time and labour.

Another noteworthy feature is the large output of products from each cell, the internal dimensions of which are about 10 × 5 × 2 feet. The two cathodes together thus have an area of about 100 square feet, and since each cell takes 2000–2500 amperes at about 5 volts, the current density on the cathodes varies from 20–25 amperes per square foot. Each cell, therefore, absorbs about 15 E.H.P., and the present installation, consisting of fifty-six cells, forty-eight of which are in constant operation, is being doubled.

Fig. 1423 shows a photographic view of the cell-room at the Electrolytic Alkali Company's works. The current efficiency of the Hargreaves & Bird cell is very high, namely, about 97 per cent. Knowing this, the E.M.F. used, and the cost of a kilowatt-hour or electrical horse-power per annum, the cost of the electrical energy required to decompose a ton of salt and to produce the corresponding amount of products can easily be calculated. Practical experience has shown that the purity of the cathode product, whether this be caustic soda or sodium carbonate, is as good as can be desired for the majority of purposes. Small quantities of impurities existing in the finished product are to a great extent independent of the electrolytic process, and have their origin in the impure carbon dioxide employed.

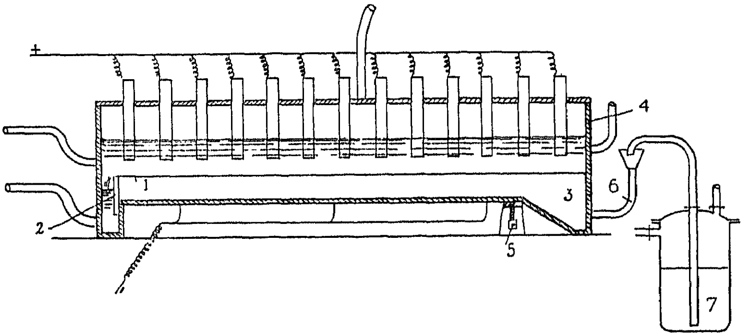


Fig. 1424.—Solvay Cell

Processes in which a Mercury Cathode is Employed.—It has long been known that sodium dissolves in mercury, producing a liquid amalgam, provided that the amount of sodium dissolved is not more than about 2 per cent. Consequently, if brine is electrolysed between a carbon anode and a mercury cathode, the discharged sodium ions dissolve in the latter; and if the concentration of the amalgam is maintained sufficiently low (say not more than one-quarter of 1 per cent) the discharged sodium will not react at all, or only very slowly, on the solution of sodium chloride. If, further, the dilute amalgam so produced is continuously removed from the sphere of action and replaced by fresh mercury, it is obvious that a current efficiency not far removed from the theoretical value could be attained. The amalgam produced can thus be decomposed by water outside the cell, yielding caustic soda of a high degree of purity and hydrogen, while the regenerated mercury can be returned to the cell to take up a further portion of sodium.

A great many cells have been designed to practically solve this apparently simple problem. Of these we can only notice two. In the Solvay process, represented diagrammatically in fig. 1424, a series of carbon anodes dip into the salt solution, the chlorine evolved being carried off by a pipe in the cover. Underlying the salt solution is a layer of mercury, which constitutes the cathode. The principle chiefly involved is the lower specific gravity of the amalgam produced. In order to remove this amalgam, which accumulates on the surface of the mercury, and which may be more or less solid, fresh mercury from a montejus or pump 7 is introduced through the pipe 6 into the well 3. The amalgam formed on the surface 1 flows over the weir 2, and is removed from the sphere of action. The flow of the amalgam in this direction may be facilitated by causing the brine to circulate through the cell in the same direction. After its removal from the cell the amalgam can be decomposed by water in a similar cell or otherwise, and the regenerated mercury returned through the pump F to perform the same cycle.

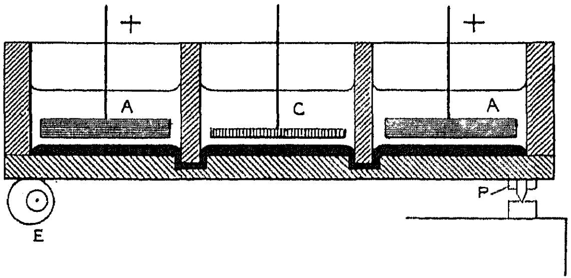


Fig. 1425.—Castner and Kellner Cell

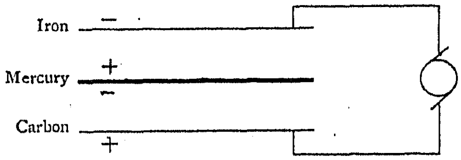
Quite different from the Solvay cell are the various cells that have been proposed by Castner and by Kellner. In this country the Castner-Kellner Company, at Weston Point, with 4000 available horse-power, operates a process based on their inventions. The process is also in operation at Niagara and elsewhere. Up to the present, however, few of the practical details have been made known. Fig. 1425 is a diagram of the cell employed. The electrolytic slate vessel, measuring 4 feet × 4 feet × 6 inches deep, is divided into three chambers by means of two non-porous, non-conducting partitions projecting downwards into the channels in the plate closing the bottom of the cell. A shallow layer of mercury covers the floor and also acts as a liquid metallic seal between the chambers. The outer compartments in which the brine is circulated contain the graphitized carbon anodes AA, while the central chamber is provided with an iron grid cathode, represented by C. At the commencement this central chamber is filled with water containing, if required, some caustic soda to render it conducting. The positive terminal of the dynamo is attached to the carbon anodes, but the negative lead may be connected either with the mercury or with the cathode C. Let us suppose it is connected with the iron cathode. Chlorine will be evolved at the anodes, while the sodium will more or less quantitatively form an amalgam with the mercury, which acts as an intermediate or double-pole electrode. The amalgam formed in the outer chambers is decomposed into mercury and caustic soda in the central chamber, and in order to accelerate the diffusion of the sodium, and so prevent the formation of a solid amalgam, the mercury is caused to oscillate between the outer and central chambers. This is accomplished by mounting the whole cell on pivots P and on an eccentric E. On slowly rotating the shaft of the latter a rocking motion is imparted to the cell.

The amalgam in the central or decomposing cell acts as anode, and the hydroxyl $(\bar{\mathrm{O}}\mathrm{H})$ anions liberated on its surface combine with the sodium (provided there is any present), forming caustic soda; while hydrogen is evolved at the iron cathode, owing to the secondary reaction between the sodium cations and the water.

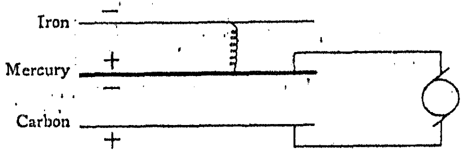
In the arrangement we are considering the same current passes in series through the brine and then through the caustic soda chamber. In actual practice the amount of sodium amalgam produced in the outer chambers never reaches the theoretical quantity. If we assume, for example, that the current efficiency, as regards amalgam production, is 90 per cent of the theoretical, then there will be a 10-per-cent deficiency of sodium to combine with the hydroxyl ions in the central compartment. The corresponding excess of hydroxyl ions, or the oxygen equivalent thereto, will therefore oxidize the mercury and render the process impracticable. To avoid this, Castner ensured the presence of sufficient sodium at the anode of the central cell by connecting the main carbon anodes A, and a lead from the mercury, with an auxiliary dynamo capable of decomposing enough sodium chloride to supply the deficiency of sodium produced in the principal circuit. He also proposed other alternatives not necessitating an auxiliary dynamo, such as electrolysing the brine for, say, one hour with the negative cable attached directly to the mercury, and then for, say, nine hours with the negative cable connected with the iron cathode. In order to make the process continuous, therefore, as much of the current as corresponds to the difference between the theoretical and the actual current efficiency in the charging compartment must be shunted off and prevented from passing through the discharging compartment.

So far we have only considered the pith of Castner's inventions. The difficulties are either that the mercury will become oxidized if an attempt is made to sail too close to the wind, or, in trying to obviate this, that the amalgam will become too rich in sodium and solidify. Both of these are overcome in Kellner's invention, which consists simply in short-circuiting the cathode C with the mercury. A galvanic couple (amalgam–caustic soda solution–iron) is thus formed, which causes a rapid solution of the sodium, but which ceases to act the moment all the sodium is dissolved. The diagram, fig. 1426, due to Engelhardt, shows the Castner and the Kellner connections of the cell and dynamo.

The current efficiency in the Castner-Kellner plants is about 90 per cent, and an E.M.F. of about 4.0–4.5 volts is employed per cell. Knowing the final products in the Castner and in the Kellner processes, and the thermochemical data, it can easily be shown, by applying Thomson's theorem, that the E.M.F. required to start the decomposition in the two cases is respectively about 2.3 and 3.2 volts. The higher E.M.F. actually used is, of course, as in the case of copper refining, to overcome the resistance of the electrolyte, and so permit the use of a higher current density and the consequent economy in plant, &c. The current density at the mercury cathode is stated to be over 100 amperes per square foot, while at the carbon anodes it is about 150 amperes per square foot. It has already been stated that sodium carbonate, and not caustic soda, is the final cathode product in the case of the Hargreaves & Bird process; and while we are referring to the E.M.F. required, according to Thomson's theorem, for the decomposition of brine, it may be pointed out that if the heat evolved during the carbonation of the caustic goes to reduce the E.M.F. which is necessary for decomposition, then only about 1.9 volt instead of 2.3 volts should produce sodium carbonate.



(Castner)



(Kellner)

Fig. 1426

Assuming a current efficiency of little short of 100 per cent, then even in the best of the processes hitherto devised the energy efficiency does not much exceed .50 per cent, if at all.

In the Castner-Kellner process the caustic soda solution of about 20 to 25 per cent NaOH is almost chemically pure. The Rhodin process differs from the Castner-Kellner process mainly in the means employed for removing the sodium amalgam formed from the sphere of action and bringing it into contact with water for the purpose of producing caustic soda and regenerating the mercury.

Processes in which Fused Salt is Electrolysed.—In the Hulin, Vautin, and Ackers processes, the latter now in operation at Niagara, molten salt is electrolysed between carbon anodes and a molten lead cathode, the sodium lead alloy being subsequently decomposed by steam, producing caustic soda, while the regenerated lead is returned to the process. Limits of space preclude any further reference to these processes, but it may be stated that the output of products for a plant of moderate dimensions is much greater than when the aqueous solutions are electrolysed. On the other hand, the E.M.F. required is much higher; but as an offset against this we have the fact that practically no evaporation of water is required.

The energy used in the decomposition of brine can easily be calculated from the preceding necessarily meagre account. Assuming that an E.M.F. of, say, 5 volts is required per cell, and that the current efficiency is 90 per cent, then 1000 kilograms (1 metric ton) of salt would require the expenditure of—

$$\frac{1000000 \times 96540 \times 5}{58.5 \times 0.9 \times 3600 \times 1000} = 2547 \text{ kilowatt-hours.}$$

With a theoretical current efficiency and an assumed theoretical E.M.F. of

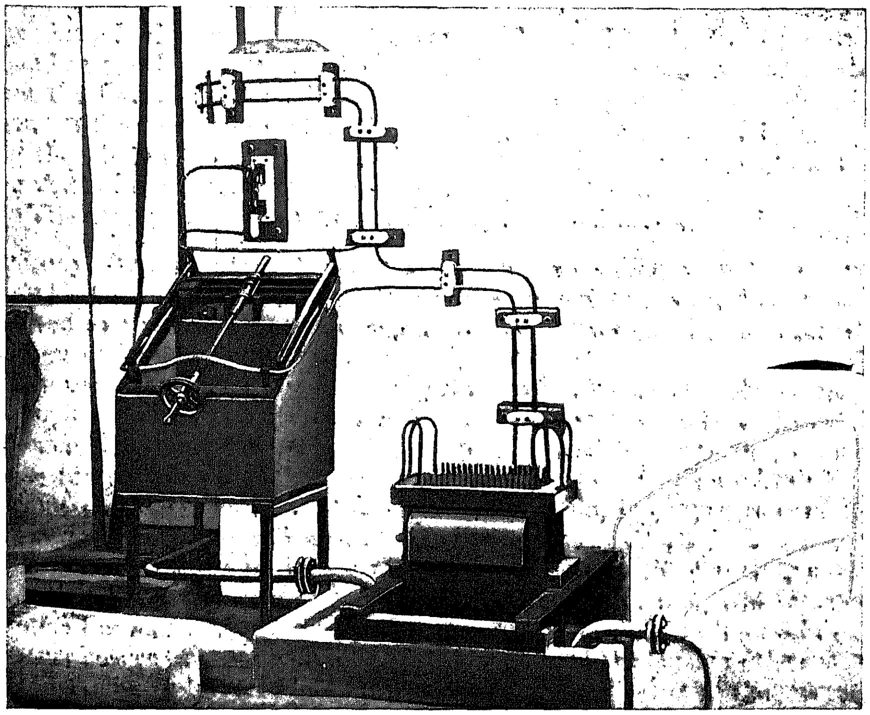


Fig. 1427.—20 Horse-power Kellner Plant (Siemens and Halske)

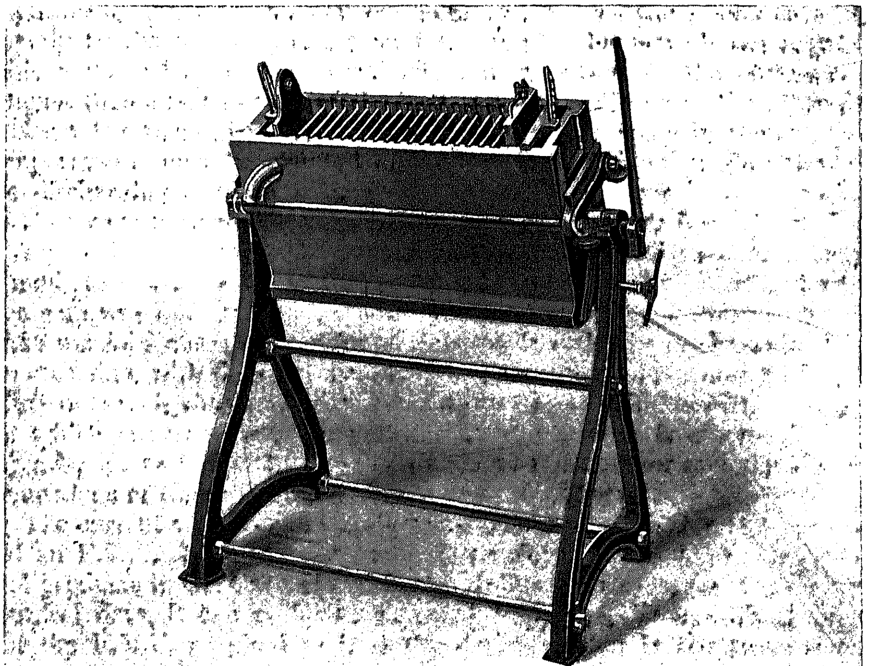


Fig. 1428.—Similar apparatus to above, but with Double-Pole Carbon Electrodes (Haas and Oettel)

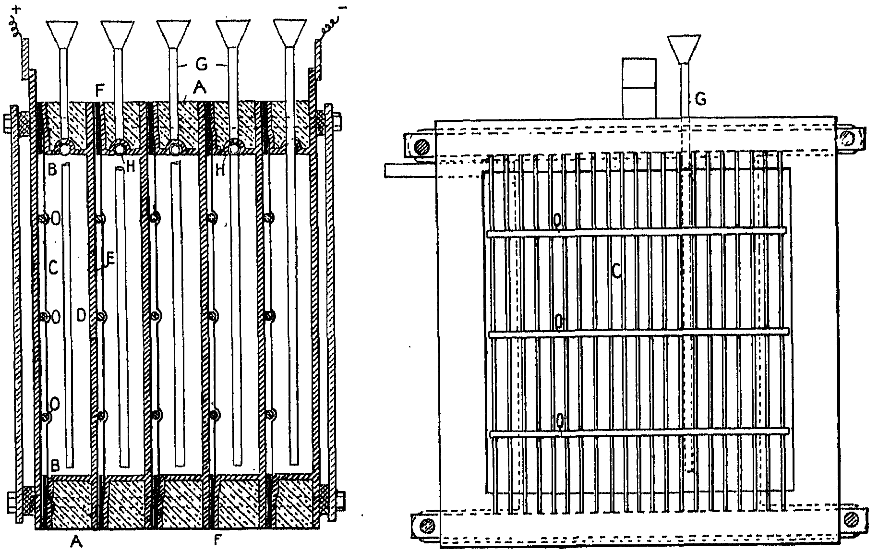
To face v, 288

2 volts per cell, only 917 kilowatt-hours would be required. Knowing the cost of the electrical energy, and the amount and value of the products obtainable from 1 metric ton of salt, the relative fraction of the manufacturing cost for electrical energy alone can easily be found. Those who care to make the calculation will find that this item, although high, is not all too serious even when the electrical energy is derived from steam-power; but limits of space prevent us from entering fully into the matter.

Hypochlorites and Chlorates.—Solutions containing calcium hypochlorite can readily be obtained by treating bleaching-powder with water. It is sometimes convenient, however, to prepare such solutions directly by the electrolysis of a solution of brine or other chloride. We have already seen that chlorine is produced at the anode, and usually caustic soda at the cathode. If these products are subsequently brought together outside the cell the reaction $2\,NaOH + Cl_2 = NaCl + NaOCl + H_2O$ takes place in the cold at a temperature not exceeding 20° C., producing hypochlorite; while at a higher temperature sodium chlorate is obtained, according to the equation $6\,NaOH + 3\,Cl_2 = 5\,NaCl + NaClO_3 + 3\,H_2O$.

Under suitable conditions the reactions can be made to take place in the cell itself, and without the intervention of a diaphragm. Bleaching solutions containing 10 grams active chlorine per litre and more can be readily obtained for an expenditure of anything between 8 and 15 E.H.P. hours per kilogram of active chlorine, depending on the conditions. The addition of a small quantity of sodium chromate favours the production ot hypochlorite. The electrodes may either be made of platinum-iridium, as in the Kellner apparatus, or of carbon, as in the Haas & Oettel plant. A much higher concentration of the bleaching chlorine is not easily obtained economically, owing to the transportation of electricity by the hypochlorite formed as well as by the unaltered salt. Such solutions are used to a considerable extent on the Continent in textile works, and in the bleaching of cellulose. The accompanying photograph, fig. 1427, shows a 20-horse-power Kellner plant installed by Siemens & Halske in a textile works, while fig. 1428 represents a similar apparatus, but with double-pole carbon electrodes, designed by Haas & Oettel.

In Europe alone 28,000 horse-power is available for the manufacture of chlorates. Formerly a diaphragm was employed, as in the Gall & Montlaur process, but in more recent processes it has been abandoned. Many types of cell are used, but that employed by the National Electrolytic Co. at Niagara Falls may be considered as an example. A vertical cross-section through a series of the cells and a side elevation are represented in fig. 1429. The faces and inside of the wooden frames A, measuring internally 18 × 26 inches, are covered with lead, while D is a lead plate covered on one side E with a thin sheet of platinum which forms the anode. A rubber gasket F, ¼ inch thick, insulates the anode E from the cathode C, and makes the series of cells water-tight when they are bolted together. The cathode is composed of vertical copper wires placed close to the anode, and prevented from short-circuiting by horizontal insulating-rods O. The lead lining of the frames places the cathode in metallic communication with the lead plate supporting the anode of the next cell. The pipes G convey the potassium chloride solution to the bottom of the cells, while H are the outlets for liquor and gas. Owing to the construction of the cell a high current density can be employed, viz. about 500 amperes per square foot. The heat generated maintains the solution at about 70° C., or the proper working temperature; the exact regulation of the temperature being effected by the rate of flow of the cold solution into the cells. No attempt is made to obtain a strong solution of chlorate, which, if formed, would reduce the current efficiency by also transporting electricity. When potassium chloride is thus electrolysed, the resulting solution containing less than 3 per cent of chlorate is cooled, when a portion of the chlorate crystallizes out. More chloride is then dissolved in the same liquor, which is again returned to the cells to perform the same cycle of operations. The current efficiency of this cell is stated to be about 70 per cent of the theoretical.



Vertical Cross Section

Side Elevation

Fig. 1429.—Cells for manufacture of Chlorates (National Electrolytic Co.)

Sodium.—This metal is now exclusively made by electrolytic methods, the chief of these being Castner's. In this process, which is similar to that originally employed by Davy for the isolation of the metal, fused caustic soda is electrolysed between iron, or more probably nickel electrodes. Fig. 1430 shows the apparatus patented by Castner. The iron containing vessel A is set in a flue, if necessary, and the temperature carefully regulated so that the molten caustic does not exceed its melting-point by more than about 20°. H is the cathode, and F the anode. The caustic in the lower portion is allowed to set, and thus seal H in position. C is a tubular iron receptacle for collecting the sodium which floats to the top, and which can be removed by a perforated ladle. There is a cylinder of iron gauze which, since it is not wetted by the sodium, prevents the latter from straying to the anode. S is asbestos or other insulating material.

An E.M.F. of 2.2 volts is sufficient to electrolyse dry fused caustic, but in practice an E.M.F. of about 5 volts is employed, and a current density of about 2000 amperes per square foot of cathode. Although apparently a very simple problem, the reactions occurring during the electrolysis of fused caustic are in reality complex.

The ions which conduct the current are $\overset{+}{\mathrm{Na}}$ and $\overset{-}{\mathrm{OH}}$. It might therefore be expected that sodium only would be obtained at the cathode, while water and oxygen would be produced at the anode, according to the equation $2\,\mathrm{OH} = \mathrm{H_2O} + \mathrm{O}$. Owing, however, to the solubility of sodium, sodium peroxide, and even water in the molten caustic, rather complicated secondary reactions take place, and influence to a very considerable extent the current efficiency. Hydrogen is nearly always produced at the cathode along with the sodium. At a temperature a few degrees above the limit mentioned no sodium at all is obtained. The Niagara Electrochemical Company are said to produce 1 lb. of sodium for an expenditure of about 4 E.H.P. hours.

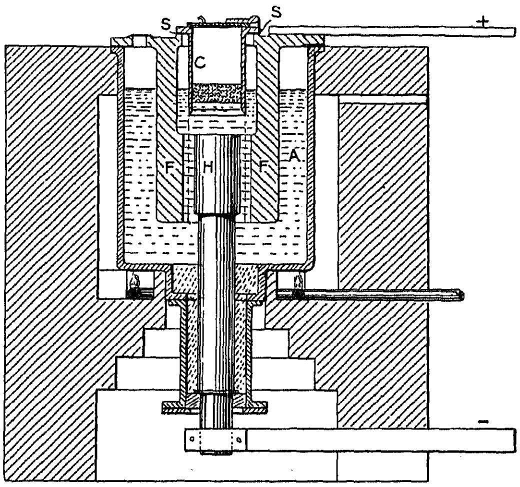


Fig. 1430.—Castner apparatus for production of sodium

Aluminium.—In the manufacture of aluminium by the electrolysis of alumina we have an illustration of the applications both of electrical heating and of electrolytic decomposition. Alumina alone is not used in practice, but a solution of 20—25 per cent $\mathrm{Al_2\,O_3}$ in a molten bath composed of cryolite, $\mathrm{Al_2\,F_6}$, 6 Na F, or some similar bath. It is generally understood that aluminium fluoride is added to the molten bath to bring its composition to about that required by the formula $\mathrm{Al_2\,F_6}$, 2 Na F. Fluorides of the metals of the alkali earths may also be employed. According to one statement a fused solvent composed of 677 parts of aluminium fluoride, 251 parts of sodium fluoride, and 234 parts of calcium fluoride has been extensively used. In all practical processes, such as those of Heroult and Hall, the bath is maintained in a molten condition at about 750°–850° C. by the development of Joule heat.

Fig. 1431, taken from a paper by Hunt (*Proc. Inst. C. E.*, 11th Feb., 1896), represents a perspective view of the electrolytic cells or pots employed by the Pittsburg Reduction Company. Chandler (*Journ. Soc. Chem. Ind.*, 1900, 609) states that these pots are rectangular iron boxes thickly lined with carbon, and having an internal cavity of 4 feet 6 inches × 2 feet 6 inches × 6 inches. The anodes consist of 40 carbons, in four rows of 10, each 3 inches in diameter and 18 inches long when new. The cell used by the British Aluminium Company at Foyers is also a carbon-lined iron box, in which the carbon blocks forming the anodes are suspended a short distance from the bottom. According to the conditions, the E.M.F. practically employed varies between 3 and 10 volts. By applying Thomson's theorem it may be shown that the E.M.F. required to decompose alumina is about 2.8 volts, which is reduced to 2.2 volts if the oxidation of the anodes is considered as part of the reaction. The greater expenditure of electrical energy in practice results in the production of heat which maintains the bath in a molten condition, so that external heating is not required. As in the case of the production of sodium, the temperature of the bath must not be allowed to rise too high, otherwise there is a danger of the aluminium becoming contaminated by carbon, and of a lower current efficiency, due to the greater solubility of the metal in the electrolyte and its reoxidation at the anodes.

The carbon lining, or the aluminium which collects on it, acts as the cathode. Since aluminium is specifically heavier than the electrolyte when both are in the fused state, it forms a pool of molten metal on the bottom, and can be siphoned off or ladled out from time to time. Wallace (*Journ. Soc. Chem. Ind.*, 1898, 310) states that at Foyers a current density of about 700 amperes per square foot of cathode surface is employed, each cell taking 8000 amperes. The area of these cells must therefore be about 11 or 12 square feet, or the same as those used by the Pittsburg Reduction Company.

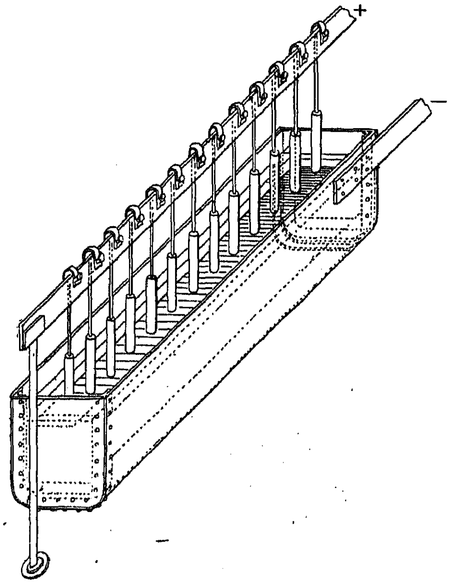


Fig. 1431.—Electrolytic Cells (Pittsburg Reduction Co.)

Fresh alumina is added from time to time to replace the amount electrolysed. The resistance of the bath increases as the alumina is consumed, and at the latter works a low voltage incandescent lamp is attached to the terminals, and indicates by glowing brightly when a fresh supply is required.

Various estimates place the amount of electrical energy required at from about 12–20 E.H.P. hours per pound of aluminium. The annual output of aluminium is now about 6000–7000 tons, and its purity is higher than formerly. This is due not so much to improvements in the electrolytic process as to the use of purer materials and purer carbon electrodes. The oxygen evolved at the anodes oxidizes the carbon to carbon monoxide, while the majority of the impurities existing in the anode, as ash, find their way into the bath and thence into the metal.

The applications of aluminium and its alloys are well known and need not be mentioned here.

Other Electrochemical Processes.—This article would be incomplete without reference to some other electrochemical processes. Among these are the electrolytic production of persulphates, perchlorates, magnesium, &c., and the electrolytic oxidation and reduction of numerous organic compounds. So far the industries referred to have fallen naturally under the headings of electrothermal or electrolytic processes. The so-called silent electrical discharge in air gives rise to the production of ozone, which appears to be employed to a limited extent. Ozonized air produced in the well-known way is now being used for the purification of water. With the exception of a few specially resistant types, all pathogenic bacteria, including cholera and typhus, are entirely destroyed by the ozone treatment. From experiments made with a plant erected at Berlin by Siemens and Halske it is estimated that the cost of treating 26,000 gallons of water with ozone is about ¾*d.* per 1000 gallons for the ozone required, or, including the cost of pumping and amortization, about 2½*d.* per 1000 gallons.

Water is decomposed into oxygen and hydrogen on a commercial scale, the gases being subsequently used for various purposes, such as autogenous soldering in secondary-battery factories, &c.

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